Development of a Dynamic Model for a Constant Tension Winch

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***Abstract*— Developing an effective controller for a towing winch is very important to maintain constant towing rope tension between a submarine cable burying machine and a cable ship. A valve-controlled hydro-motor system for the towing winch is a highly nonlinear and time varying system due to the flow–pressure dynamics, oil leakage, oil temperature variation, etc. In this paper, we develop the dynamic model of the valve-controlled hydraulic winch by linearizing its nonlinear dynamics at an operating point. Both the simulation and experimental results demonstrate that the developed model describes the dynamic relationship between input control signals from the electronic proportional valve and pull forces generated by hydro-motor well. We used the model to evaluate and analyze the PID and FUZZY P+ID controllers for the constant tension winch that operates under sea waves with significant variations (peak-to-peak) from 1.5 to 2.5 meters.**

Keywords—constant tension winch; valve-controlled hydraulic motor; active controller

I. INTRODUCTION

The exploitation of oceanic resources increases the demand on operations of burying electric cables in seabed. A very challenging task in ocean engineering is to control a submarine to bury cables effectively and reliably. A typical burying machine for burying submarine cables is a cable burying ship equipped with a towing winch. Fig. 1 shows a submarine cable burying machine and a constant tension winch connected by a towing rope. The winch has to use its constant tension control system to maintain the towing rope tension within a predetermined range when driving the cable burying machine,

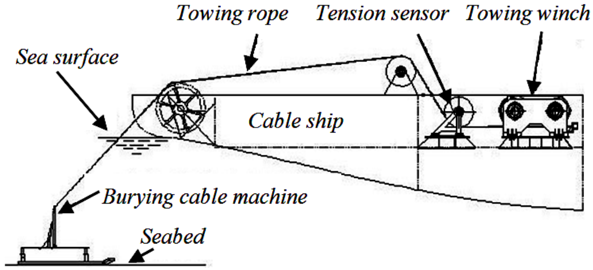

Fig. 1. The towing winch in the sea cable burying system

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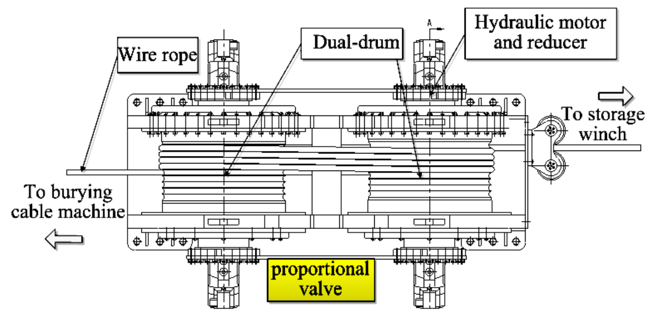

Fig. 2. The towing winch

i.e., the towing winch pays in or out the towing rope to regulate its tension value when the actual tension exceeds the predetermined tension range, because too high towing rope tension causes the cable burying machine to be overturned or too low tension can make the towing rope to be intertwined with an umbilical cable or a submarine communication cable. Especially, cable ship movements and wave-forces cause considerable inertia and drag undetectable forces on the towing rope, so it is very difficult to keep the constant tension during operating the cable burying machine with ship movements and in adverse sea conditions with significant wave variations.

A constant tension winch controller can divide into a passive or an active type. The work [1] presented a typical passive constant tension winch controller with a hydraulic system, which operates via an adjustable relief valve. The passive controller delivers low precision in sea environments with uncertainty and needs to estimate an actual load for adjustment. Another key issue is that the amount of cooling was required for the hydraulic system if the hydraulic pump was working at full power consumption. In this paper, we discuss a constant tension winch with an active controller that regulates a valve-controlled hydraulic motor. However, conducting sea trials to test the active controller via the real winch is a very cost and time-consuming task. Therefore, before control algorithms for the active controller are implemented on the winch system, simulation studies of the control algorithms must be done to evaluate its performance. For simlation studies, the mathematical models for describing dynamic characteristics or behavior of physical system must be established [2, 3]. In this paper, we develop an analytical model of the valve-controlled hydraulic winch for representing dynamic relationships between input signals from the electronic proportional valve and pull forces generated by the hydro-motor.

II. STRUCTURE OF CONSTANT TENSION TOWING WINCH

The constant towing winch shown in Fig. 2 is a dual-drum one driven by four valve-controlled hydraulic motors. There are two types of hydraulic motor control systems: the valve-controlled motor system and the pump-controlled motor system. The valve-controlled motor system has faster dynamic responses than the pump-controlled motor system does. Because the constant tension winch should be able to change rotation direction and speed promotly, the valve-controlled hydro-motor system was chosen to control the winch. The hydraulic system consists of a hydraulic pump unit to supply hydraulic fluid to the system components, and the four hydro-motors to drive the dual-drum, i.e., two motors to control a drum. The primary component of the active controller is an electronic controlled proportional directional valve which regulates the flow of hydraulic fluid into the hydraulic motors, and thus controls the speed and the rotation direction of the motors. The flow through the valve is approximately proportional to an input control signal value, i.e., increasing or decreasing the input signal value allows the towing winch to pay in or to pay out the towing rope.

III. MATHEMATIC MODEL OF HYDRAULIC SYSTEM

Being similar to other hydraulic systems, the valve-controlled hydro-motor system is a highly nonlinear and time varying system due to the flow–pressure dynamics, oil leakage, oil temperature variation, etc. We develop the transfer functions of the valve-controlled hydraulic motor by linearizing its nonlinear dynamics at an operating point.

The general equation for load flow through an ideal proportional directional valve is expressed by[4]

$$q_L = C_d \omega x_v \sqrt{\frac{1}{\rho}\left(p_s - \frac{x_v}{|x_v|} p_L\right)}, \tag{1}$$

where q_L is the load flow, C_d is the discharge coefficient, ω is the area gradient, x_v is the valve spool position, ρ is the fluid mass density, p_s is the load pressure, and p_L is the supply pressure.

Obvioursly, Eq (1) for the valve-controlled hydraulic system is a highly nonlinear. In order to develope its transfer functions, we linearize Eq (1) by a Taylor's series around a particular operating point $q_L = f(x_{vA}, p_{LA})$ as follows

$$q_L = q_A + \frac{\partial q_L}{\partial x_v}\Big|_A \Delta x_v + \frac{\partial q_L}{\partial x_L}\Big|_A \Delta P_L + \cdots, \tag{2}$$

The higher order infinitesimals can be neglected when the control range is limited to the vicinity of the operating point, so Eq (1) becomes

$$q_L = q_{LA} = \Delta q_L = \frac{\partial q_L}{\partial x_v}\Big|_A \Delta x_v + \frac{\partial q_L}{\partial x_L}\Big|_A \Delta P_L, \tag{3}$$

and the linear equation for the steady-flow through the proportional directional valve is approximated as

$$\Delta q_L = K_q \Delta x_v + K_c \Delta p_L, \tag{4}$$

where Δq_L is an incremental value of q_L, and Δx_v is an incremental value of x_v that represents the displacement of the proportional directional valve, Δp_L is an incremental value of p_L that represents the pressure of the payload. The flow gain, K_q, represents an incremental value of load flow caused by the displacement of valve spool as follows

$$K_q = \frac{\partial q_L}{\partial x_L} = C_d \omega \sqrt{\frac{1}{\rho}(p_s - p_L)}. \tag{5}$$

The flow-pressure coefficient, K_c, indicates an increment value of the load flow caused by variation of load pressure as follows

$$K_c = -\frac{\partial q_L}{\partial P_L} = 2 C_d \omega x_v \sqrt{\frac{1}{\rho}(p_s - p_L)}. \tag{6}$$

The fluid flow equation of the hydraulic motor is expressed by

$$q_L = D_m \frac{d\theta_m}{dt} + C_{tm} p_L + \frac{V_t}{4\beta_e} \frac{dp_L}{dt}, \tag{7}$$

where D_m is the volume displacement rate of the hydraulic motor with respect to its shaft rotation, θ_m is the rotation angle of the hydraulic motor, C_{tm} is the leakage coefficient, V_t is the compressed volume, and β_e is the effective bulk modulus indicating the hydraulic fluid compressibility. The torque equation of the motor and load is expressed by

$$D_m p_L = J_t \frac{d^2\theta_m}{dt} + B_m \frac{d\theta_m}{dt} + G\theta_m + T_L, \tag{8}$$

where J_t is the motor plus load inertia with respect to the motor shaft, B_m is the equivalent viscous damping coefficient of motor plus load, G is the elastic stiffness of load, and T_L is the load torque with respect to the motor shaft.

The three Laplace transform functions of the valve-controlled hydraulic motor are obtained from (4), (7), and (8) respectively,

$$Q_L = K_q X_v - K_c P_L, \tag{9}$$

$$Q_L = D_m s\theta_m + C_{tm} P_L + \frac{V_t}{4\beta_e} s P_L, \tag{10}$$

$$P_L D_m = J_t s^2 \theta_m + B_m s\theta_m + G\theta_m + T_L. \tag{11}$$

The viscous damping and elasticity loads are neglected because their impacts on the system are quite small compared with other external disturbance forces. Combining (9)-(11), the open-loop transfer functions of the motor shaft angular rate response for the incremental value x_v and external load torque T_L become

$$G(s)_1=\frac{\dot{\theta}_m(s)}{X_v(s)}=(\frac{K_q}{D_m})/(\frac{s^2}{\omega_h^2}+\frac{2\xi_h}{\omega_h}+1), \tag{12}$$

$$G(s)_2=\frac{\dot{\theta}_m(s)}{T_L(s)}=\frac{K_{ce}}{D_m^2}(1+\frac{V_t}{4K_{ce}\beta_e}s)/(\frac{s^2}{\omega_h^2}+\frac{2\xi_h}{\omega_h}+1), \tag{13}$$

where $K_{ce}= K_c+ C_{tm}$ is total pressure-flow coefficient, ω_h and ξ_h are the natural frequency and the damping ratio of the hydraulic system

$$\omega_h=\sqrt{\frac{4\beta_e D_m^2}{J_t V_t}}, \tag{14}$$

$$\xi_h=\frac{K_{ce}}{D_m}\sqrt{\frac{\beta_e J_t}{V_t}}. \tag{15}$$

The relationship between the input of proportion valve U_e and the incremental value X_v is

$$X_v=K_e U_e, \tag{16}$$

where K_e is the proportional coefficient. Defining I as the reduction ratio of the reducer, we have the relationship between the shaft angle rate of the winch $\dot{\theta}_j$ and the rotation angle rate of the hydraulic motor $\dot{\theta}_m$

$$\dot{\theta}_j=\dot{\theta}_m/I. \tag{17}$$

The external load torque T_L is

$$T_L=FR, \tag{18}$$

where F is the towing rope tension, and R is the winch drum radius.

We develop the block diagram of the valve-controlled hydraulic motor system, as shown in Fig. 3, from the equations of the hydraulic system. The input to the model is the control signal of the electronic proportional valve ranged in [-10V, +10V]. The output from the model is the rotation angular velocity of the winch. The load torque T_L is treated as an external disturbance.

IV. SIMULATION AND RESULTS

To verify the effectiveness of the hydraulic winch mode developed in Section III, we conduct the simulation studies of the winch model with no load and 3 ton load, respectively, and compare the simulation results with the experiment data.

For simulation studies, we use parameters' values of the valve-controlled hydraulic motor listed in Table I. The values of I, R, D_m and K_e are determined from manufacturer data, while the values of β_e, and V_t are estimated according to designed data, and the values of K_q, K_{ce}, J_t, ω_h, ξ_h, are derived by calculation. Among these parameters, the total pressure-flow coefficient, K_{ce}, is state-varying, so it is the main factor resulting in the hydraulic system nonlinearity. In this paper, we choose K_{ce}=8.5×10^{-11} for the simulation studies.

TABLE I. PARAMETERS OF THE HYDRAULIC SYSTEM MODEL

Parameter	Value
K_q	7.2×10^{-2}m^3/s
K_{ce}	[1.5×10^{-11}m^2, 8.5×10^{-11}m^2]
K_e	1.5mm/V
I	95.83
β_e	700MPa
D_m	7.96×10^{-6}m^3/rad
V_t	9.6×10^{-3}m^3
J_t	1.25×10^{-2}kg·m^2
ω_h	38.45
ξ_h	0.33
R	426mm

Plugging the parameters from Table I into the transfer functions $G(s)_1$ and $G(s)_2$, we obtain

$$G(s)_1=\frac{1.34\times10^7}{s^2+25s+1470}, \tag{19}$$

$$G(s)_2=\frac{1.31s+0.05}{s^2+25s+1470}. \tag{20}$$

We use Matlab/Simulink to simulate the step response of the winch model as follows.

1) The step response of the model with no load. Under the condition without load, no external disturbance needs to be considered in the transfer functions, as shown in the block diagram of Fig. 4. Giving the model inputs with the step voltages of 5V and 10V respectively, we plot the step time responeses of the simulation results in Fig. 5.

2) The step response of the model with 3 ton load. Generally, the tension on the towing rope is set to 3 ton. Under the condition of 3 ton load, the external disturbance has to be considered in the transfer functions, as shown in the block diagram of Fig. 6. Giving the model inputs with the step voltages of 5V and 10V respectively, we plot the step time responses of the simulation results in Fig. 7.

Table II shows that the comparative stdudy of the step responses of the simulation model and the experiment data from the in-water test on the winch.

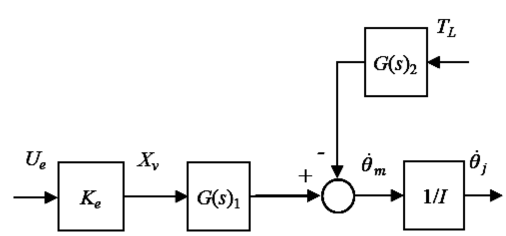

Fig. 3. Transfer function block diagram of the winch hydraulic system

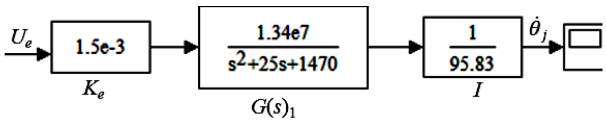

Fig. 4. Block diagram of the winch model with on load

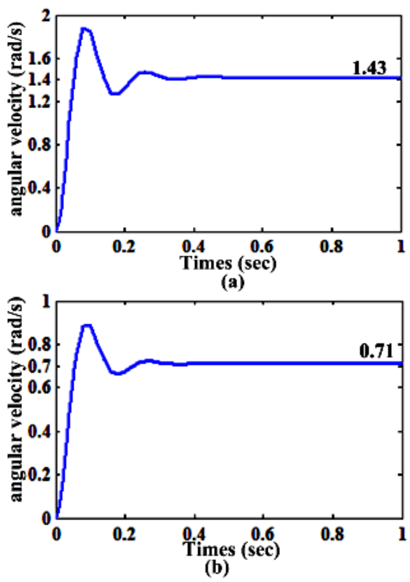

Fig. 5. The results of the simulation with no load: (a) with 10 v as the input signal, (b) with 5 v as the input signal

TABLE II. SIMULATION RESULTS AND EXPERIMENT DATA WITH NO LOAD AND 3 TON LOAD

Input signal	load	Simulation result	Experiment result
5V	no load	0.71	0.7
5V	3 ton	0.7	0.68
10V	no load	1.43	1.42
10V	3 ton	1.42	1.4

V. CONCLUSION

This paper presents a model for an active controlled constant tension winch. The structure of the winch is constructed based on valve controlled hydraulic motor. To verify the effectiveness of the model, the simulation of the winch model is developed.

Both the simulation studies and the in-water test results listed in Table II match very well, so we used the model to evaluate and analyze the PID and FUZZYP + ID controllers [5][7] for the constant tension winch that operates under sea waves with significant variations (peak-to-peak) from 1.5 to 2.5 meters in [8].

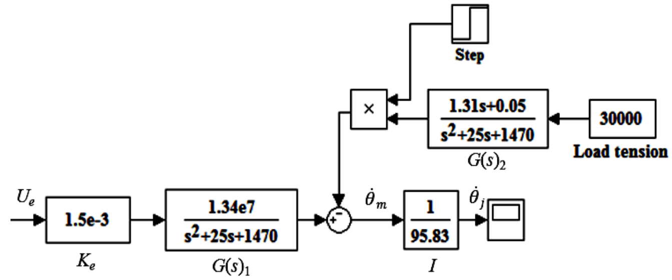

Fig. 6. Block diagram of the winch model with 3 ton load

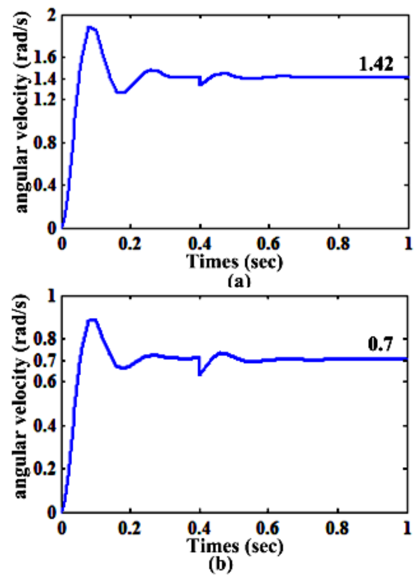

Fig. 7 The results of the simulation with 3 ton load: (a) with 10 v as the input signal, (b) with 5 v as the input signal

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