Numerical simulation and control loop optimization of the electric heave compensation system for the ship crane

K V Chupina, A M Khannanov, V K Usoltsev

Far Eastern Federal University, 8, Sukhanova St., Vladivostok, 690091, Russia

E-mail: kirachupina@yandex.ru

Abstract. The paper is devoted to a problem of synthesis of the automatic electric drive for the heave compensating mechanism of the ship crane. The main disturbance for the drive of the compensating mechanism is load torque caused by ship’s heave and sea waves. Ship’s heave and sea waves are stochastic processes. For the synthesis of the automatic control system of the electric drive, the identification problem for main disturbance was solved. The designed mathematical model of sea waves allows one to synthesize the automatic system with the modal controller, observer and hybrid feedback-feedforward architecture. As result, vertical displacements of a towed underwater vehicle have been reduced hundreds of times.

1. Introduction

There are many ship electric automated systems which operate in conditions of stochastic external disturbances. These are propulsive complexes, steering complexes, ship systems of dynamic positioning, towing and trawl winches, lifting systems, towed and remotely operating vehicles during launch and recovery operations using ship cranes. The main type of stochastic external disturbances for these systems is sea waves.

Sea disturbance can be a reason of propeller’s emergence. That is why the propeller thrust and the ship velocity decrease, the rudder-propeller interaction and the course keeping become more difficult. The vertical displacements of the head block of the hoisting device caused by ship heave motions take place when using towed and remotely operating vehicles. The dynamic strains caused by these displacements are forced to cable with the underwater vehicle and change its immersing depth. As a result some underwater operations become impossible or are considerably complicated [1, 2].

So the successful use of underwater vehicles depends on the actions that are taken for the reduction of the influence of the ship heave on the deviation of the underwater vehicle’s immersion depth. For this purpose ship hoisting devices should have special compensating systems. The drive of such systems should operate in the mode of heave compensation of the underwater vehicle’s immersion depth [1, 2].

The main requirements for automatic control systems of the compensating mechanism’s electric drive are stability, fast response time, effective compensation of the main disturbance, optimal transient responses to reference-input signals.

2. The selection of the electric drive of the compensating mechanism

It is expedient to choose the brushless direct current motor as the drive of the compensating mechanism. During heave compensation the drive operates in the reversing duty. The brushless direct
current motor can develop electromagnetic torque, which will be sufficient for overcoming the load torque even at a full stop of a shaft. Speed-torque and speed-current curves of the brushless direct current motor are completely similar to curves of separately excited direct current motors or permanent magnet motors.

The brushless direct current motors have some advantages in comparison with other types of drives. The first is the main specific weight-dimensional parameters as the power-to-weight ratio (kW/kg) and the torque-to-weight ratio (kNm/kg); the brushless direct current motors outclass a direct current and induction motors. The second is high torque capacity (pick torque and current capacities can exceed nominal rates five and more times). The third is high energy efficiency (energy conversion efficiency above 90%); fast response time. The fourth is almost a zero slope of the speed-torque characteristic (in the closed-loop system with the speed controller) and almost an unlimited controlled speed range (1:10000 and more). The last is a long life, high reliability and extending operation life due to the lack of the sliding electric contacts.

The mathematical model of the brushless direct current motor is based on a standard system of the differential equations [3].

3. The mathematical model of the main disturbance

The main disturbance for the drive of the damping device is load torque caused by the vessel’s heave and sea waves.

Now the mathematical description of the behavior of ship devices, mechanisms and systems in the frequency domain is traditionally used for designing these elements. These are spectra and frequency responses. For the considered complex, such description is insufficient since it doesn't provide required accuracy. Therefore it is necessary to consider the behavior of the object which is forced by sea waves, in a time domain. That is, for the synthesis of the automatic control system of the electric drive, it is necessary that the mathematical model of the main disturbance was suitable for calculations using a system of the differential equations.

The process of sea waves is chaotic: subsequent waves differ in shape, amplitude and period. Such process is irregular, it can be described as a stochastic. Since stages of generation, growth, propagation and decay of sea waves take place, this process is stochastic non-stationary. For the range from 20 minutes to twelve hours sea can be considered as stationary ergodic process. Such approach considerably simplifies the mathematical description of both the sea waves, and results of its force experienced by different objects, in particular, of the vessel heave [4].

For the mathematical description of irregular sea waves the 2D model is chosen. The sea state is described by the wave directed to the marine object and wave spectra. The spectrum which has been accepted at the 12-th International Towing Tank Conference is used as a basis for simulating sea waves.

The techniques for receiving the rational approximating functions of a wave spectrum have been developed. The received mathematical models give a possibility to analyze processes in a time domain with the necessary accuracy and to calculate width spectra and the average peak period of the sea wave state.

Simulations were produced in MATLAB. The offered approximations by rational functions give a possibility to use standard blocks from Simulink Library Browser for creation of the model block diagram. The «white noise» signal passes through a filter with the rational transfer function for the simulation main disturbance. The block diagram is shown in fig. 1.

The offered model can be used for calculation of the sea wave disturbance by any forces. The sea code is set by the corresponding values of the variance “D” and the peak wave frequency “wm”.

The calculated rated wave spectra and the rated correlation function are shown in fig. 2. The symbol “x” is the frequency rated to the peak wave frequency; symbol “t” – time; s and R – spectrum and correlation related to the variance of waves corresponding. It is obvious that the results obtained using the developed mathematic model are well consistent with the experimental research.
Figure 1. The mathematical model of the sea wave disturbance

Figure 2. The rated wave spectra $s(x)$ and the rated correlation function $R(t)$ are calculated in the time domain (thin curves) and in the frequency domain [4] (thick curves)

In order to receive an absolute value of the spectrum function, it is necessary to multiply ordinates by the square of the Gain block value (fig. 1), but abscissa – to peak the wave frequency. In order to receive the absolute value of the correlation function, it is necessary to multiply ordinates by the variance of waves.

Simulations results and comparison versus the data of the international sea scale (Douglas scale) are provided in table 1 to check the validity of the model. Parameters were calculated for the high height values.

### Table 1. Dependence of the average height of waves on the sea state code

| Waves average height | 1  | 2  | 3  | 4  | 5  | 6  | 7  | 8  |
|----------------------|----|----|----|----|----|----|----|----|
| Douglas scale data, m| 0-0.1 | 0.1-0.5 | 0.5-1.25 | 1.25-2.5 | 2.5-4 | 4-6 | 6-9 | 9-14 |
| Model data, m        | 0.095 | 0.49 | 1.244 | 2.49 | 3.92 | 5.98 | 8.55 | 13.78 |

It is shown that the proposed model provides high accuracy of calculations and can be used to synthesize the observer and the controller for the electric drive.
4. Synthesis of the electric drive’s automatic control system of the compensating mechanism

Due to the solution of the disturbance identification problem for the calculation electric drive’s automatic control system, it is possible to design the MIMO (Multi-Input Multi-Output) system with the modal controller and the hybrid feedback-feedforward architecture [5, 6]. For this purpose the plant model was expanded by the combination of mathematical models of the electric drive and sea waves as the main disturbance. State variables of the expanded plant (electric drive and disturbance models) are inputs for the modal controller.

For astatic control, it is necessary to expand the electric drive model of an integral of the motor shaft output angle [1]. Thus, EMF, current, speed, a shaft output angle, integral of the shaft output angle, and the reflected motor shaft speed, acceleration, two derivatives of accelerations and angular movement of the head block of the damping device caused by disturbance are components of a state vector. Some components of a state vector can be used as measured feedback, the others – as observed feedback. The reference motor shaft angle and a signal “white noise” are plant model inputs. The block diagram of the electric drive’s automatic control system is shown in fig. 3.

![Figure 3. The block diagram of the hybrid feedback-feedforward architecture of the electric drive’s automatic control system: the brushless direct current motor (Subsystem), disturbance load (Subsystem1), controller (Subsystem2)](image)

The strategy of linear quadratic optimal control was used to calculate gain components of the modal controller. The synthesis of the astatic Lyuemberg observer has been executed to estimate the components of the expanded plant’s state vector. The observer’s modal controller has been synthesized so that the place of desired closed-loop poles provided faster transients than plant’s ones.
did. The calculation has shown that the observer settle time is eight times less than the plant’s settle time.

Standard deviations of underwater vehicle’s vertical displacements were calculated for estimation of the efficiency of the automatic control system for heave compensation. Simulating was carried out using the switched-on and switched-off compensating mechanism’s drive. With the sea state code “four-five”, these calculated values are \((9.2-9.8) \cdot 10^{-3}\) m and \(0.588-0.843\) m respectively. The calculation has shown that the designed compensating mechanism’s drive reduces vertical displacements of a vehicle more than 600 times.

Actually the compensating nevertheless will not be so ideal because in the real system there are some nonlinearity and uncertainty, for example, dry and elastic friction, variations of cables tension and moment of inertia of the compensating mechanism, sensor errors, noises and others. These variations have not been considered during system design. Also the compensating error substantially depends on the accuracy of estimating the state variables received in the observer and sensors. Respectively, the less difference among estimates, real values, and the sensor error, the more precisely the automatic control system will compensate the main disturbance.

The transients for the reference position of the damping device’s head block are shown in fig. 4. Calculations were carried out for cases of the step reference signal and slope with the unlimited reference signal.

![Figure 4. Drive speed transients of the compensating mechanism: 1 – for the step reference signal, 2 - for the slope with the up limited reference signal](image)

It is visible that, when using the slope reference signal, overshoot has decreased ten times, from 9% to 0.9%. The speed oscillations of the compensating mechanism’s drive are excluded when using the slope reference signal.

5. Conclusion
As a result of research, the hybrid feedback-feedforward architecture of the automatic control MIMO system with the modal controller which has allowed one to receive desirable dynamics of the electric drive has been synthesized.

Optimization of parameters of the modal controller is realized. The synthesis of the astatic observer has been executed for obtaining information about state variables of the electric drive and the sea waves disturbance.

Standard deviations of vertical displacements of the descent-rise device head block were calculated for estimation of the efficiency of the automatic control system for heave compensation. Simulating was carried out using the switched-on and switched-off damping device drive. Calculation has shown that the designed automatic control system of the damping device reduces vertical displacements of the vehicle more than 600 times.

The developed mathematical model provides high accuracy of calculations of sea wave parameters. It can be used for designing other ship electric automated systems which operate in conditions of sea waves disturbances.
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