Xu-Yang Cao and Shan-Chao Liu*

Design of a Semi-active Heave Compensation System Combined Variable Frequency with Throttle

Abstract: In order to reduce the impact of wave to the ship crane, complete the lifting operations safely and efficiently, and with the view to decreasing the energy loss and increasing the efficiency of the existing system, a semi-active heave compensation system combined variable frequency with throttle is designed. By using the frequency conversion hydraulic technology and retaining valve control unit, the system can meet the requirements of position precision and energy consumption at the same time. Then the simulation models of the hydraulic system and the variable frequency motor are established respectively via the AMESim and Simulink. The compensation performance and energy saving effect of the system are analysed by joint simulation. The results show that 99.7% impact of the ship heave motion can be compensated. The energy consumption is only 25.6% of the valve-control active heave compensation system under the same operation conditions.

Keywords: heave compensation; variable frequency control; power matching; energy saving; joint simulation

1 Introduction

Cranes working on the sea is affected by the heave motion of the ship, which has a negative effect on hoisting. Heave compensation system is essential for the safe and efficient hoisting operation. According to the differences of energy source, the existing heave compensation system can be divided into passive system, active system and semi-active system [1, 2]. The energy consumed by the passive system stems from the heave motion of the ship, without additional power. Meanwhile, the passive system has disadvantages of large lag and low compensation precision. The active system is completely powered by the hydraulic power unit, so it has the characteristics of low hysteresis, high compensation precision, but huge energy consumption. The semi-active system, which combines the advantages of the above two

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systems, can use the hydraulic power unit and the accumulator as power supply. And the compensation precision and system energy consumption are taken into account simultaneously.

Heave compensation technology has been comparatively mature in some western countries. In 1990s, the application of semi-active heave compensation system to the floating offshore drilling platform has been proposed in the United States [3]. Based on the secondary regulation of hydrostatic transmission, a new type of heave compensation system was designed and manufactured by Rexroth research. The compensation efficiency of this system can reach 95%, and 65% of the load gravity potential energy can be recycled [4]. In this field, some achievements have been made in recent years in China [5, 6]. Most of the researches have been focused on how to improve the compensation performance of heave compensation system. Comparatively, researches of energy conservation are less.

As a global power matching transmission mode, the variable frequency hydraulic speed regulation technology has been widely used in many fields because of its high efficiency and good energy saving effect. These advantages of the technology are more obvious in the case of large power and large flow fluctuation. For example, the world’s first attempt to apply the variable frequency hydraulic technology to the hydraulic elevator was taken place in the Mitsubishi in Japan [7]. Another example is the application of variable frequency drive principle in injection moulding machine which was developed and tested by Helbig A [8].

In this paper, the frequency conversion technology is applied to heave compensation system and a new type of semi active heave compensation system is designed. Then the compensation and energy saving effectiveness of this system is verified by joint simulation. A new idea and method for the design and energy saving of heave compensation system are put forward.

2 System Principle Design

2.1 Analysis of the Ship and Compensation Motions

According to the relevant literature, the trajectory of ocean wave is a kind of non-linear random curve [9]. It can be seen as a superposition of sine waves of different periods, amplitudes and phases. It can be roughly defined as

$$X_b = \sum_{n=1}^{N} \frac{H_n}{2} \sin \left( \frac{2\pi}{T_n} t + \theta_n \right)$$

(1)
As the wave height curve is generally similar to the shape of the sine curve, in order to facilitate the initial analysis and design, the wave motion can be approximated as a sinusoidal motion, which can be described by the following equations

\[
x_b = \frac{H}{2} \sin \left( \frac{2\pi}{T} t \right)
\]  
(2)

The frequency of ship’s heave motion is the same as that of wave motion, and the amplitude is smaller. The equation of the ship’s heave motion is

\[
x_c = \frac{\mu H}{2} \sin \left( \frac{2\pi}{T} t \right)
\]  
(3)

When the pulley block magnification is 1, in order to make the load in the suspended state is not affected by the heave motion of the ship, the compensation motion should content the following equation

\[
x_g = -\frac{\mu H}{4} \sin \left( \frac{2\pi}{T} t \right)
\]  
(4)

In the above equations, \(T\) is the wave period, \(H\) is the wave height, the \(\theta\) is the initial phase angle, and the \(\mu\) is the ratio of the ship’s heave displacement to the wave height.

### 2.2 Principle of the Variable Frequency Hydraulic Speed Regulation

Variable frequency hydraulic speed regulation technology is a kind of global energy-saving driving mode, which combines the frequency conversion technology and hydraulic transmission [10]. AC variable frequency motor drives hydraulic quantitative pump to output flow required by the system. Then the motor can be always in high efficiency. The output flow of hydraulic pump is defined as

\[
Q_v = V_p n \eta_v = \frac{60 V_p n \eta_v f (1 - s)}{p}
\]  
(5)

Where \(V_p\), \(n\) and \(\eta_v\) are hydraulic pump displacement, input speed and volume efficiency. \(f\), \(s\) and \(p\) are the motor’s power frequency, slip ratio and the number of poles.
Variable frequency hydraulic speed regulation system has no overflow loss and throttling loss, so it has higher power efficiency. Compared with the traditional variable displacement speed control mode, it also has the advantages of simple structure, low noise, large speed range, convenient control and better energy saving, etc. Meanwhile, there are some disadvantages such as slow dynamic response, poor low speed characteristics, and low speed accuracy. The vector control inverter is a good solution for these problems. Vector control simulates asynchronous motor to DC motor by coordinate transformation, and realizes the decoupling control of flux and torque, so that AC motor can achieve the control effect of DC motor.

### 2.3 System Principle Design

The semi-active heave compensation system combined variable frequency with throttle, as shown in Fig. 1, consists of passive compensation part and active compensation part. The passive compensation part formed by cavity C and an accumulator 13, which is equivalent to a hydraulic spring, bears most of the weight of the load. The active compensation part is comprised of cavity D, cavity E, servo valve 8, inverter 3, motor 4, pump 5, etc. Its function is to overcome the load inertia and friction. The Ship heave displacement signal x1 and piston rod displacement signal x2 are both detected by the sensor and used as the input of the controller. The combined action of solenoid directional valve 12, servo valve 8 and inverter 3, which are controlled and coordinated by the controller, makes the piston rod complete the compensation accurately.

This system not only retains the advantages of the valve control system, such as high control precision, fast response, but also allows full play to the characteristics of variable frequency hydraulic speed control system, which contains large speed range and good energy-saving. On the one hand, the difference between the pump outlet pressure and the working pressure of the hydraulic cylinder is kept constant under the control of the solenoid directional valve 12 and the pressure compensation valve 6. Thus, the flow into the actuator is only determined by the servo valve 8 spool size and the servo valve 8 is driven by the control signal to realize the accurate and fast control of the cylinder 9. On the other hand, in order to achieve energy saving and high efficiency, a variety of measures are applied to the system. The energy consumption of active compensation part is greatly reduced because the passive compensation part supports the load. The solenoid valve 12 and the pressure compensation valve 6 can be used to fulfill the pressure matching of the system. The real-time required flow of the system is calculated according to the servo valve control signal. Then the inverter motor according to the results of this calculation to adjust the output flow of the pump is adjusted by the inverter motor to achieve the system flow matching.
The working process of the system is divided into ascending compensation and descending compensation. When the controller 11 receives the rising signal of the ship, it drives the servo valve 8 to move up. The hydraulic oil flows from cavity D pass the solenoid valve 12 into the spring chamber of pressure compensation valve 6. At the same time, the control signal is sent out by analysing the displacement signals x1 and x2. Servo valve 8 moves toward right and spool opening size is controlled. Frequency conversion motor 4 and constant pump 5 provide enough flow for the system. The high pressure oil enters the cavity D through the servo valve 8 to push the piston rod to retract. Wire rope is released to achieve load displacement compensation. The process of descending compensation is similar and it's not described in this article.


Fig. 1: Schematic diagram of hydraulic system
2.4 Controller Design

The heave compensation control mode can be divided into displacement compensation, velocity compensation and tension compensation according to the purpose of compensation. A control method combining velocity compensation and displacement compensation is designed, as shown in Fig. 2, and it enables the system to compensate random waves efficiently. According to Eq. 3 and Eq. 4, the relative velocity of the piston rod and the cylinder body should be 1/2 of the ship’s heave velocity in opposite direction. The expected velocity is used as the instruction signal of the speed controller which is open loop. In this way, the hydraulic cylinder can respond in time to make the compensation action. The difference between the ship heave displacement and the relative displacement of piston rod and cylinder body is used as the input of the displacement controller to form a closed loop control. Then, the cumulative error of velocity compensation is eliminated and the accuracy and stability of the system are guaranteed.

![Fig. 2: Block diagram of the control principle](image)

3 Modeling and Simulation

There is an independent model of the speed controlled motor in AMESim, but the model is too idealistic, which will reduce the reliability of the simulation results. Therefore, after establishing AMESim model of hydraulic system and Simulink model of vector variable frequency motor, the whole system is analysed by using the joint simulation, which can give full play to the advantages of two kinds of software in their respective fields.

3.1 Hydraulic System Modeling

According to the schematic diagram, the hydraulic system simulation model, as shown in Fig. 3(a), is established. During the modelling, the influence of pressure on
the viscosity and elasticity of oil is not considered; the elasticity of the steel wire rope is neglected and efficiencies of hydraulic cylinders and pump is not counted.
3.2 Vector Variable Frequency Motor Modeling

The simulation model of vector frequency conversion motor is shown in Fig. 3(b). The model is composed of motor module, inverter, DC power supply, motor measurement unit, hysteresis pulse generator, flux observer, coordinate conversion, flux regulator, torque regulator, speed regulator and other sub modules. Coordinate conversion module can be established according to the mathematical model, or replaced by the ready-made modules, which are abc_dq0, Transformation and dq0_abc Transformation. Flux regulator, torque regulator and speed regulator are modelled as the PI regulator.
4 Simulation Analysis

Ships equipped with this heave compensation system should be able to work under degree 4 sea situations, which mean the maximum compensation height is 3 meters and the minimum compensation period is 5 seconds. A load of 10 tons is chosen for simulation. The main parameters of the simulation model are calculated, as shown in Table 1.

**Tab. 1:** Main parameters of amesim simulation model

<table>
<thead>
<tr>
<th>title</th>
<th>unit</th>
<th>value</th>
<th>title</th>
<th>unit</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load quality</td>
<td>m/t</td>
<td>10</td>
<td>Pump displacement</td>
<td>ml/r</td>
<td>80</td>
</tr>
<tr>
<td>Passive cavity working area</td>
<td>mm²</td>
<td>11684</td>
<td>Rated flow of servo valve (Δp=30bar)</td>
<td>L/min</td>
<td>200</td>
</tr>
<tr>
<td>Cavity D working area</td>
<td>mm²</td>
<td>8858</td>
<td>Accumulator volume</td>
<td>L</td>
<td>200</td>
</tr>
<tr>
<td>Cavity E working area</td>
<td>mm²</td>
<td>8858</td>
<td>Accumulator precharge pressure</td>
<td>MPa</td>
<td>17.1</td>
</tr>
<tr>
<td>Maximum stroke of compound cylinder</td>
<td>mm</td>
<td>500</td>
<td>Accumulator initial pressure</td>
<td>MPa</td>
<td>17.5</td>
</tr>
</tbody>
</table>

4.1 Analysis of Harmonic Input

The harmonic signal generated by multiple sinusoidal signals is used as the input of the system to simulate the heave motion of the ship under the action of random waves. These sinusoidal signals’ amplitudes range from 0.05 to 0.25 meter and their periods range from 5 to 15 seconds. The displacement curves of ship and load are shown in Fig. 4(a). From this figure, the ship heave displacement amplitude is 0.8 meter, and Load displacement amplitude is 2.5 millimetres. The displacement compensation efficiency of the system is 99.7%, which is very ideal.

Accumulator volume and pressure curves are shown in Fig. 4(b). The accumulator pressure fluctuates with the changes of ship’s heave and the accumulator volume is changed reversely. When the ship rises up, the piston of the composite cylinder moves down, the oil in the passive chamber is pressed into the accumulator, the pressure of the accumulator is increased and the volume is reduced. When the ship descends down, the piston of the compound cylinder moves up, the oil are entered the passive cavity from the accumulator, the pressure of the accumulator is reduced, and the volume is increased.

According to the pressure change curve of the servo valve, as shown in Fig. 4(c), port P pressure is always 3MPa higher than the pressure of port A or port B. In other words, the pump output pressure is well matched with the load pressure. The max-
The minimum pressure of the port P of the servo valve is 9MPa and the pressure variation range is about 3MPa, which are very beneficial to stability of the hydraulic system. The cause of the lower system pressure and smaller pressure fluctuation range are that the high pressure oil provided by the pump is only used to overcome the inertia and friction of the system.

The flow tracking curve of system is shown in Fig. 4(d) and the speed tracking curve of vector motor is shown in Fig. 4(e). It can be seen that the vector control motor meets the speed control requirements of the system and drives the pump to provide just enough flow, though there is a little lag between the output speed and the desired speed. The average flow rate of overflow is 11.9L/min, while the output flow rate of the pump is about 56.6L/min, which ensures a high efficiency of the system. There are two main reasons for the overflow loss. The first reason is that the output flow of the pump is set to be slightly larger than that required by the hydraulic cylinder. The other reason is that the motor speed should not be lower than the minimum speed, which is 300r/min in this system, to ensure the normal operation of the pump. When the required flow rate is less than 24L/min, the pump is driven by the motor at the minimum speed and the system is only controlled by the valve.
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(b) Accumulator pressure [bar] and accumulator volume [L]

(c) Pressure at port P [bar], pressure at port A [bar], pressure at port B [bar], and pressure at port T [bar] over time.
Fig. 4: (a) Displacement curves of ship and load. (b) Curves of accumulator volume and pressure. (c) Pressure fluctuation of the servo valve oil ports. (d) Flow tracking curve of system. (e) Speed tracking curve of vector motor.
4.2 Comparison of System Energy Consumption

In order to compare the energy consumption of the above system and the existing active and semi-active heave compensation system, the simulation models of the other three systems are established: valve-control active system (Fig. 5(a)), active system combined variable frequency with throttle (Fig. 5(b)) and valve-control semi-active system (Fig. 5(c)).

The sine wave signal, whose amplitude is 0.45 meter and period is 5 seconds, is used as a unified simulation input signal. Simulation time is set to 30 seconds. The energy consumption of the system can be calculated by the equation

\[ E = \int p_p q_p \, dt \]  

(6)

Where the pump output pressure is \( p_p \) and the pump output flow is \( q_p \). The energy consumptions of the four systems can be easily obtained by using the post-processing function of AMESim. The results of the initial 5 seconds were ignored to eliminate the impact of the instability at the beginning of the simulation on the analysis. Comparison data are shown in Table 2.

As shown in Table 2, the system designed in this paper has an obvious advantage in energy saving. Its energy consumption is only 25.6% of that of the valve-control active system, or 45% of that of the active system combined variable frequency with throttle, or 53.1% of that of the valve-control semi-active system.
**Fig. 5:** (a) The simulation model of valve-control active system. (b) The simulation model of active system combined variable frequency with throttle. (c) The simulation model of valve-control semi-active system

**Tab. 2:** Comparison of energy consumption of four systems

<table>
<thead>
<tr>
<th>system name</th>
<th>energy consumption in 5 to 30 seconds (kJ)</th>
<th>average energy consumption of a cycle (kJ)</th>
</tr>
</thead>
<tbody>
<tr>
<td>valve-control active system</td>
<td>1809.5</td>
<td>361.9</td>
</tr>
<tr>
<td>active system combined variable frequency with throttle</td>
<td>1033.1</td>
<td>206.6</td>
</tr>
<tr>
<td>valve-control semi-active system</td>
<td>873.5</td>
<td>174.7</td>
</tr>
<tr>
<td>semi-active system combined variable frequency with throttle</td>
<td>464.2</td>
<td>92.8</td>
</tr>
</tbody>
</table>

**5 Conclusion**

A semi active heave compensation system is designed by combining the existing heave compensation system with the variable frequency hydraulic technology. The
system energy consumption is greatly reduced under the premise of ensuring the high compensation precision. The hydraulic system model and vector motor model are established by AMESim and Simulink respectively.

Under the precise control of the controller and the servo valve, the compensation efficiency of the system for random heave motion of ship is increased to 99.7%. The weight of the load is born by the “hydraulic spring” composed of the passive cavity of composite cylinder and the accumulator. Pressure and flow matching are achieved simultaneously by active compensation part and the efficiency of the whole system, including the motor, is improved. The energy saving advantage of this system is verified by comparing the energy consumptions of four different heave compensation systems. When the experimental conditions are ripe, a prototype should be built according to this paper to verify the correctness of the simulation results and to lay the foundation for the batch production of the corresponding products.

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References