Study on Energy Recovery Performance of Linear Active Heave Compensation System

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Abstract. In order to verify the feasibility of the energy recovery of the linear active heave compensation device under the descent compensation condition, the author established a schematic diagram of the hydraulic system based on the heave compensation principle, performed a simulation analysis of the hydraulic system model in a cycle, analysed energy loss of the system under specific sea conditions. Therefore, it is concluded that the gravity potential energy of the load under the descent compensation condition has great potential for recovery. Based on the results of the demonstration, the article proposes an electro-hydraulic energy recovery system of the linear active heave compensation device based on a hydraulic motor-generator. The system model is simulated and analysed in a single cycle under standard operating conditions. The results show that: When the system parameters match reasonably, the heave compensation device has a higher compensation accuracy. The energy recovery rate of the system under the down compensation condition reaches 21.6%, and the effect of energy saving is great. At the same time, simulation analysis shows that under the premise of ensuring the compensation accuracy, changes in the quantitative motor displacement and the viscous friction coefficient of the generator will affect the energy recovery efficiency to varying degrees.

1. Introduction

With the construction of a maritime power and the increasing frequency of ocean exploration, China's ocean operations such as deep-sea exploration are also increasing, but the marine environment is very bad. In order to reduce the impact of the movement in the direction of the heave on the safe operation at sea, various types of marine equipment usually use heave compensation devices during engineering operations [1, 2]. Facing the increasingly severe environmental status quo and the problem of energy shortage, according to the requirements of energy saving and environmental protection, the linear active heave compensation device has attracted the attention of industry and academia due to its energy recovery potential [3, 4]. Recently, it can be seen that the related research on linear active heave compensation devices [5, 6] mainly focuses on the control algorithm analysis [7, 8] and the design of
hydraulic system-related actuators [9, 10], and the research on energy recovery related aspects has not yet been involved. According to the research, it is learned that the research theory of energy recovery system is very rich in engineering machinery such as hydraulic excavators and hydraulic forklifts [11, 12], and has strong practicality. The core components of its hydraulic system are based on hydraulic cylinders [13, 14], so they can be compared and analyzed in research and application, as a theoretical reference for the application of hydraulic energy recovery systems in linear heave compensation devices [15].

The energy that can be recovered by the linear active heave compensation device is mainly the gravitational potential energy during the descent compensation process. In the existing devices, the energy generated by them is eventually lost in the form of thermal energy, and few studies have been conducted on the recovery of the gravity potential energy during the down compensation of the linear heave compensation device. Under its complete working cycle, the linear active heave compensation device can be regarded as rising compensation and falling compensation. The time of a single working cycle is about 10s, and the energy of gravity potential energy can be recovered during the falling compensation phase, about 5s. Based on the existing basis, the feasibility of energy recovery is simulated and analyzed. The author proposes an electro-hydraulic energy recovery system of the linear active heave compensation device based on a hydraulic motor-generator to realize the energy recovery of the load's gravity potential energy during the down compensation phase.

2. Composition and working principle of linear active heave compensation system

As shown in Fig.1, the linear active heave compensation device is mainly consist of offshore platform 1, load 2, movable pulley 3, active compensation hydraulic cylinder4, fixed pulley 5, servo valve 6, relief valve 7, pump 8, motor 9, tank 10.

![Figure 1. Schematic of the linear active heave compensation device.](image)

2.1. Working principle of linear active heave compensation system

It can be seen from Fig. 1 that the linear active heave compensation device completes a heave compensation process in one working cycle, and the piston rod of the active heave compensation hydraulic cylinder completes the extension movement and the contraction movement. In order to reduce the influence of the heave direction when hoisting loads on an offshore platform, heave compensation devices must be activated.

The dynamic equation of the motion of the active heave compensation hydraulic cylinder is:

\[ m_a \ddot{x}_a = P_2 A_2 - P_1 A_1 + F_a \]  

\[ F_a = 2m_a g + F_d + B_a \dot{x}_a + f_a + d \]  

(1)

(2)
In which, \( m_a \) is the equivalent mass included in the load of the system; \( x_a \) is the movement displacement of the piston rod of the active hydraulic cylinder; \( P_1 \) is the pressure of the rod cavity of the active hydraulic cylinder; \( A_1 \) is the area of the rod cavity of the active hydraulic cylinder; \( P_2 \) is the pressure of the active hydraulic cylinder with a rod cavity; \( A_2 \) is the active area of the active hydraulic cylinder with a rod cavity; \( F_a \) is the total external load force; \( F_d \) is the drag force of the water body that the load may be subjected to in the water; \( B_a \) is the active hydraulic cylinder piston linear motion viscous damping coefficient; \( f_a \) is the linear motion friction force of the piston of the active hydraulic cylinder; \( d \) is other uncalculated and unmodeled forces.

The flow continuity equation of the active hydraulic cylinder is:

\[
\frac{V_{01} + A_1 x_a}{\beta_e} \dot{P}_1 = A_1 x_a - C_t (P_1 - P_2)
\]  

(3)

\[
\frac{V_{02} - A_2 x_a}{\beta_e} \dot{P}_2 = A_2 x_a - Q_2 + C_t (P_1 - P_2)
\]  

(4)

In which, \( V_{01} \) is the sum of the initial volume of the active hydraulic cylinder rodless cavity and the pipe connecting the rodless cavity; \( V_{02} \) is the sum of the initial volume of the active hydraulic cylinder with the rod cavity and the pipeline connected to the rod cavity; \( Q_1 \) is the active hydraulic cylinder without the amount of hydraulic oil flowing in the rod cavity; \( Q_2 \) is the amount of hydraulic oil flowing in the rod cavity of the active hydraulic cylinder; \( C_t \) is the total internal leakage coefficient of the hydraulic cylinder due to the pressure difference; \( \beta_e \) is the effective volume of hydraulic oil in the hydraulic system elastic modulus.

Servo valve flow equation:

\[
Q_v = C_{d} W x_v \sqrt{\frac{2 \Delta p}{\rho}}
\]  

(5)

In which, \( Q_v \) is the flow rate through the servo valve, which is determined by the displacement and time of the active cylinder; \( C_{d} \) is the flow coefficient; \( W \) is the flow area gradient of the servo valve; \( x_v \) is the spool displacement; \( \Delta p \) is the servo valve pressure difference; \( \rho \) is fluid density of hydraulic oil.

In combination with Fig. 1, the working principle of heave compensation stage is described as follows:

1) Rise compensation stage. When the hull moves downward under the action of waves, in order to keep the relative position of the load unchanged, the load needs to move upward relative to the hull, at this time, the heave compensation device needs to be enabled. According to the working principle diagram in Fig. 1, the piston rod of active hydraulic cylinder 4 needs to be extended at this time. In this stage, when the control center detects the downward movement of the hull through the displacement sensor, it will send out the command of upward compensation. The hydraulic oil source provides the pressure. Through the electro-hydraulic servo valve 6, the oil source supplies oil to the rodless cavity of the active hydraulic cylinder. The piston rod is forced to move upward to extend, and the load 2 moves upward under the traction to complete the upward compensation.

2) Descent compensation stage. When the platform moves upward due to the influence of waves, in order to keep the relative position of the load unchanged, the load needs to move downward relative to the hull. At this time, the heave compensation device is enabled. According to the schematic diagram in Fig. 1, the piston rod of the active heave compensation hydraulic cylinder needs to contract at this time. In this stage, when the control center detects the upward movement of the hull through the displacement sensor, it will issue the command of downward compensation. At this time, the piston rod of the active hydraulic cylinder is affected by the weight of the pulley 3 on the piston rod and the force transmitted by the load 2 through the cable, which is the gravity effect of the external load. Under the action of external force, the hydraulic oil in the cavity without rod is under pressure. The oil flows through the
electro-hydraulic servo valve 6 and finally to the oil tank 10. The heave compensation system uses the electro-hydraulic servo valve to control the downward movement speed of the piston rod of the cylinder.

2.2. System simulation and energy loss analysis

The feasibility of energy recovery of the linear active heave compensation device is analyzed and verified by the author. According to the schematic diagram in Fig. 1, the author establishes an AMESim simulation analysis model of the linear active heave compensation device without energy recovery system, as shown in Fig. 2, which analyzes the system energy loss in a heave compensation period under the three-level sea conditions. In this study, the main active heave compensation device for energy recovery is valve controlled cylinder system. In the valve control cylinder system model of this study, the core component is mainly the hydraulic cylinder and its corresponding control valve. In the study, the designed total external load mass is 3T, the piston diameter of the active heave compensation hydraulic cylinder is set as 70mm, the piston rod diameter is set as 45mm, and the stroke is set as 2000mm. When the heave compensation device works, the large cavity pressure of the active compensation hydraulic cylinder given by experience is: the load pressure $P_1$ is 200 bar, the wave condition is set as three-stage sea condition, the maximum frequency is 0.1Hz, and the wave amplitude is 1m.

![Simulation model of the linear active heave compensation device](image)

**Figure 2.** Simulation model of the linear active heave compensation device.

In order to study the energy loss of the device in a cycle, the simulation experiment model is shown in Fig. 2. The simulation results analyze the pressure and flow of the return pipeline without rod cavity in the process of lowering compensation, and calculate the recoverable power and energy of the system.
According to Fig. 3, the compensation device has better compensation effect. From the displacement simulation curve, it can be seen that in a compensation cycle, the descent compensation stage of the working condition, as shown in Fig. 3, has the conditions of gravity potential energy recovery within the total length of 5S from 2.5s to 7.5S, and then it is analyzed.

Figure 4. Flowrate of return pipeline.

Figure 5. Pressure of the system.

It is known that the descent compensation phase is from 2.5s to 7.5S of the working cycle. According to Fig. 4 and Fig. 5, it can be seen that the pressure of the oil return pipeline without rod chamber is about 150 bar at this stage, while the system oil source pressure is 0 at this time. The device drives the hydraulic cylinder under the action of the load gravity, and the maximum flow of the oil return pipeline is about 140L / min.

According to the simulation results and system principle, in the process of descent compensation, because the system uses servo valve to control the lowering speed of the hydraulic cylinder, the gravity potential energy of the load is all converted into heat energy consumption, which is transmitted to the oil to make its temperature rise. In this process, according to the simulation results, the power integration can be obtained, and the energy loss of the system in the whole descent compensation stage is about 118.4KJ. It can be seen from the working principle that this part of energy comes from the gravitational potential energy of the external load. Through the simulation experiment of the linear heave compensation device, this part of energy can be recovered, so the linear heave compensation device with energy recovery system can be designed.
3. Working principle and analysis of linear heave compensation device with energy recovery system

3.1. Working principle

This research is about the energy recovery of the piston rod of the active hydraulic cylinder under the action of the load gravity, so the influence of the hydraulic oil loss and other external effects is not considered. As shown in Fig. 6, according to the basic principle of heave compensation, the motion state schematic diagram of the active hydraulic cylinder in the process of simulation heave compensation is designed. The energy recovery system consists of electromagnetic reversing valve 10, check valve 11, hydraulic motor 12, and generator 13.

![Figure 6. Linear active heave compensation device with energy recovery system.](image)

When the load drops, the recoverable gravitational potential energy due to the height drop:

\[ E = mg \times H \]  

(6)

In which, \( E \) is the weight potential energy (kJ) that can be recovered when the load moves in compensation; \( m \) is the mass of the load (T); \( g \) is the gravity acceleration of the environment (N / kg); \( H \) is the height of the load falling (m).

For the inlet of the energy recovery hydraulic motor, the flow continuity equation of the hydraulic oil can be expressed as follows:

\[ \frac{V_3}{\beta_p} \dot{P}_3 = Q_v - \frac{\omega_m D_m}{2\pi} - C_m P_3 \]  

(7)

In which, \( Q_v \) is the flow into the mouth of the energy recovery motor, and the oil supply pressure of the hydraulic system is in this energy recovery stage \( Q_v = Q_1 \); \( V_3 \) is the volume into the mouth of the energy recovery hydraulic motor; \( \omega_m \) is the angular speed of the energy recovery hydraulic motor; \( D_m \) is the theoretical displacement of the energy recovery hydraulic motor; \( C_m \) is the total leakage coefficient of the energy recovery hydraulic motor.

The dynamic equations of rotor motion on the shaft of hydraulic motor and generator are as follows:

\[ J_m \frac{d\omega_m}{dt} = \frac{P_m D_m}{2\pi} - T_m \]  

(8)
In which, $J_m$ is the total moment of inertia of the rotor of the energy recovery system hydraulic motor generator combination unit; $T_m$ is the total torque of the energy recovery system hydraulic motor generator combination unit.

In this paper, the experimental schematic diagram of linear heave compensation device with energy recovery system is analysed.

(1) Rise compensation stage. At this stage, the solenoid directional valve (10) is in the left position, and the hydraulic oil of the system returns to the oil tank. The working process is the same as that of the original system. The hydraulic oil supply pressure makes the piston rod of the active hydraulic cylinder extend to achieve the purpose of rising compensation.

(2) Descent compensation stage. In this stage, the energy recovery system is designed into the heave compensation device. Its working process is as follows: after the control centre sends the descent compensation signal, the electromagnetic reversing valve (10) is in the right position, and the hydraulic oil source stops supplying pressure. Under the effect of the external load gravity, the piston rod of the active heave compensation hydraulic cylinder shrinks under the external force, and the speed of the piston rod in the descent compensation process is controlled by the servo valve (6) to achieve the purpose of compensation. In the process of piston lowering of the active hydraulic cylinder, the released gravitational potential energy is recovered by the hydraulic motor (12) and converted into kinetic energy. Finally, the hydraulic motor drives the generator (13) to operate and generate electricity to complete the energy recovery of the system.

3.2. System simulation and energy recovery analysis
According to the hydraulic schematic diagram of linear active heave compensation device with energy recovery system, AMESim software is used to establish the experimental simulation model, as shown in Fig. 7. The simulation model ignores the friction coefficient of hydraulic motor and generator, the volume and mechanical efficiency of motor and pump. The signal of control centre is replaced by the original signal library. The parameters of simulation model are shown in Table. 1.

![Simulation model of the linear active heave compensation device with energy recovery system](image)

Figure 7. Simulation model of the linear active heave compensation device with energy recovery system.
From the above simulation model, the simulation results are analysed:

(1) Analysis of heave compensation performance. According to the compensation movement of the load in a cycle, the dynamic curve of the piston rod position of the active hydraulic cylinder can be obtained from the simulation results. Fig. 8 is the position tracking simulation curve. It can be seen from the figure that the linear heave compensation device with heave compensation system has a better compensation effect. On the premise of not affecting the compensation accuracy, the next step is to study its energy recovery performance.

| Table 1. System simulation parameters |
|--------------------------------------|
| Parameter name | Parameter value |
| Stroke of active hydraulic cylinder (mm) | 2000 |
| Piston diameter of active hydraulic cylinder (mm) | 70 |
| Diameter of piston rod of active hydraulic cylinder (mm) | 45 |
| Maximum flow of servo valve path (L/min) | 80 |
| Servo valve recovery path pressure drop (bar) | 1 |
| Displacement of hydraulic motor (mL/r) | 150 |
| Rated speed of hydraulic motor (r/min) | 1000 |

Figure 8. Displacement following curve.

Figure 9. System recovery energy curve

(2) Energy recovery performance analysis. According to the principle and simulation results of the energy recovery system of the linear active heave compensation device, the energy recovery can be carried out according to the potential energy in the period of 2.5s to 7.5s in one working cycle. In the descent compensation stage, the pressure at the oil outlet of the rodless chamber of the active hydraulic cylinder and the pressure at the inlet of the hydraulic motor of the energy recovery system are shown in Fig. 9. According to the simulation results, it can be known that the pressure in the rodless chamber is about 150bar in the lowering compensation stage. The high-pressure oil output by the active hydraulic cylinder is depressurized and regulated by the servo valve, and a certain amount of pressure loss is generated, so the pressure flowing into the inlet of the hydraulic motor, the force is about 40 bar. According to Fig. 10, it can be seen that after the start of energy recovery, the flow of the energy recovery system gradually increases and reaches a stable value, the maximum return oil flow is about 145L/min, and the flow at the end of energy recovery gradually decreases and finally reaches 0.

The simulation analysis of the descent compensation stage mainly focuses on the energy recovery of potential energy. When the load falls, the pressure of the hydraulic system will rise accordingly. After the servo valve is depressurized and adjusted, the high-pressure oil will drive the hydraulic motor to work, and the hydraulic motor will drive the generator to work. After the recovery of the load heavy potential energy, it will be converted into the electrical energy in the super capacitor to complete the
energy recovery process of potential energy. The recoverable energy of hydraulic system is calculated by formula (6), and the recovered energy of hydraulic motor generator is calculated by formula (10):

\[ E = \int_{t_1}^{t_2} T_m(t) W_m(t) dt \]  

(9)

![Energy recovery system](image1)

**Figure 10.** Return oil flowrate of system.

![Energy recovery system](image2)

**Figure 11.** System recovery energy curve.

In the process of descent compensation, the gravity potential energy in the process of load lowering is recovered completely, without overflow loss, but there will be energy loss of servo valve during speed regulation and the influence of motor and load torque, which makes the energy recovered by the system far lower than the gravity potential energy of load, which is the problem to be solved in the future research. As a part of energy is consumed in the system, it can be seen from Fig. 11 that the recovered energy of the system is about 26.03KJ, and the recoverable gravitational potential energy in the descent compensation stage is 120KJ, with an energy recovery efficiency of 21.6%.

3.3. The influence of parameter change on energy recovery efficiency

The energy recovery system of the linear heave compensation device has high energy recovery efficiency. Next, the influence of the parameters of the quantitative motor and generator on the energy recovery efficiency is studied.

![Energy recovery system](image3)

**Figure 12.** Energy recovery of hydraulic motor with different displacement.

![Energy recovery system](image4)

**Figure 13.** Energy recovery of generator with different viscous friction coefficient.
Under the condition of keeping other parameters unchanged, carry out energy recovery experiments on quantitative motors with different displacement. Take 120mL/r, 150mL/r, 180mL/r and 200mL/r for motor displacement, and take the same compensation accuracy of the heave compensation device. The energy recovery performance of the system is shown in Fig. 12.

According to the above simulation results, it can be seen that under the condition that other parameters do not change, with the gradual increase of the displacement of the quantitative motor, under the same compensation accuracy, the pressure of the hydraulic motor into the mouth gradually decreases with the increase of the displacement of the motor, and the energy recovered by the energy recovery system gradually decreases, that is, the recovery efficiency of the energy recovery system gradually decreases from 33.7% down to 12.2%.

Under the condition of keeping other parameters unchanged, change the viscous friction coefficient of the generator, take 0.08Nm/(r/min), 0.1Nm/(r/min), 0.12Nm/(r/min) and 0.1Nm/(r/min). When the compensation accuracy of the heave compensation device is the same, the energy recovery of the system is shown in Fig. 13.

According to the above simulation results, it can be seen that with the increase of the viscous friction coefficient of the generator and the same compensation accuracy of the heave compensation device, the pressure into the mouth of the hydraulic motor increases with the increase of the displacement of the motor, and the energy recovered by the energy recovery system increases, that is, the energy recovery system The efficiency gradually increased from 17.4% to 30.2%.

4. Conclusion

(1) This paper analyses the composition and working principle of the linear active heave compensation device, and designs the simulation model according to its working principle diagram. Through the analysis of the system oil source pressure, the pressure of hydraulic cylinder without rod cavity and the hydraulic flow of oil return pipeline in the process of lowering compensation of the device, it is concluded that the gravity potential energy of the load weight is completely lost in this stage in the form of heat energy, so as to verify the feasibility of energy recovery of the device.

(2) According to the basic principles of energy recovery and heave compensation, a linear active heave compensation device with energy recovery system is designed, and a simulation model is built. The simulation results show that the performance of the new system meets the working requirements of heave compensation, and the compensation accuracy is high. The energy recovery efficiency of the energy recovery system of the quantitative hydraulic motor generator with the displacement of 150mL/r is about 21.6%, and the energy saving effect is good.

(3) The author makes a simulation analysis of the influence of the displacement of the quantitative motor and the viscous friction coefficient of the generator on the energy recovery efficiency of the energy recovery system. Under the condition of ensuring the same compensation accuracy of the heave compensation device, the larger the displacement of the motor, the less the energy recovered by the energy system, and the lower the energy recovery efficiency. The larger the viscous friction coefficient is, the more energy the energy recovery system recovers, and the higher the energy recovery efficiency is.

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