A New Energy Recovery Device by Utilizing the Merchant Ship Rolling

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ABSTRACT Wave energy is a renewable clean and resources for ships. According to the rolling characteristics of merchant ships, this research designs an energy recovery device installed on the ship deck to harvest the energy generated by waves. And a typical 50,000-ton ship is selected as the experimental platform of the device. The mathematical model is established and solved by Newmark-$\beta$ method, and a simulation platform is built in MATLAB. By simulating the movement of the slider in the device, it shows that the slider can realize reciprocating motion when the ship rolls continuously. Then, the relevant parameters in different combinations of natural frequency $\omega_n$ and damping ratio $\epsilon$ are calculated, the power output of the device and the motion law of the slider are analyzed, and parameters’ variable curves are drawn in figures. A reasonable combination of $\omega_n = 1.225$ Hz, $\epsilon = 0.0612$ and a relatively ideal average power (174.11 W) are found by comparing amounts of simulated results from typical sea conditions. It is indicated that four main factors affecting the power generation of the device include the elasticity coefficient $K$, effective damping coefficient $C$ in the device, the ship rolling period and rolling angle. Besides, the device can still output power continuously by simulation in complex sea condition. Based on the research, it is proved that the device mentioned in this study has good feasibility in generating electric energy during ship rolling, which can provide valuable ideas for the application of wave energy in ships on voyage.

INDEX TERMS Energy recovery device, newmark-$\beta$ method, ship, the rolling motion of ship, wave energy.

NOMENCLATURE

\begin{align*}
\omega_n &= \sqrt{\frac{K}{m}} & \text{Natural frequency} \\
\epsilon &= \frac{C}{(2\sqrt{m*K})} & \text{Damping ratio} \\
\eta &= \frac{K}{m} & \text{Transmission efficiency}
\end{align*}

I. INTRODUCTION

With the rapid development of the world shipping market, the number of merchant ships is increasing rapidly and the competition in the shipping market is becoming fierce, which caused the more and more fuel oil is consumed. IMO (International Maritime Organization) puts forward higher requirements for the discharge of ships and the concept of green ships [1], [2]. In this case, how to develop new energy resources are particularly important. On the one hand, clean energy such as wind energy and solar energy has been tested and applied in large merchant ships of COSCO SHIPPING [3]–[5], on the other hand, wave energy is also a good available resource with high energy density, large storage and wide distribution.
There are many kinds of wave and current energy utilization devices in the world, such as:

In Ref. [6], a one-based multi-buoy offshore wave energy converter (FWEC) named ‘Sharp Eagle Wanshan’ has been invented by Songwei Sheng et al. Two novel schemes of Floating WECs’ mooring system are studied in [7] and [8]. Zhang et al. [9] have analyzed an oscillating buoy type wave energy converter (WEC) by numerical simulation. Tay and Wei [10] reported the pontoon-type WEC, the WEC comprises of several interconnected floating modules which are connected by line hinges. Qi Xie et al. in [11] proposed a novel oscillating buoy WEC based on a spatial double X-shaped mechanism for self-powered sensors in sea-crossing bridges. FWECs have good effect in experiments, but these FWECs are close to the shore and unable to move with ships at any time. Wan et al. [12] developed a new WEC suitable for wave conditions in Zhoushan sea area. Yang et al. [13], Wang et al. [14], Kolios et al. [15], Zhang et al. [16], and Rahmati and Aggidis [17] studied the point absorption wave energy converter. Among them, a floating array-point-raft wave energy converter has been designed and experimented by Jimei University, the transmission mode of WEC is very instructive; reliability assessment of point-absorber wave energy has been converted by Cranfield University and the reliability assessment framework has been developed; an adaptive bistable power capture mechanism to a point absorber WEC has been proposed by Shanghai Jiao Tong University. These inventors are innovative for us and we have a better understanding of the point-absorber WECs and they are experience with installation experiments. In Ref. [18], a shore wave energy converter is studied by Halil Ibrahim Yamaç, Ahmet Koca. About the device of using current, Zhang et al. [19]–[21] developed a device to utilize the current energy. Layout optimization of landing gears for an underwater glider has been researched by Northwestern Polytechnical University; Baoshou Zhang et al. has done numerical simulation of VIV energy harvesting, the hydrodynamic energy has a good utilization value. Galvan-Pozos and Ocampo-Torres [22] presented the WEC based on the Stewart-Gough platform, a novel six-degree of freedom WEC is proposed. Furthermore, Coe et al. [23] summarized the general stages and workflow for wave energy converter design. Kolios et al. [24], Doyle et al. [25], and Liu et al. [26] evaluated and improved the efficiency of some wave energy recovery devices. On the device of rolling motion, Wang and Yu [27] studied energy converts the energy device by the rolling motion of a sailing boat into electrical energy. The device is installed inside the boat and rolling energy conversion of a boat using an eccentric rotor revolving in a hula-hoop motion, the device can sail with the ship to provide energy for the ship, this approach is interesting. Wenjun et al. [28] developed a wave energy power generation system based on an unmanned underwater vehicle. This device uses rolling motion of unmanned underwater vehicle to achieve energy conversion, they also have been designed the permanent magnet generator of the device.

At present, wave energy utilization is mainly concentrated on the offshore platforms near the coast and the immovable ships like pontoons, if a ship can use the wave energy on the voyage, it will supply power for ship electronic equipment and save ship energy consumption.

Ship often encounters bad weather when it is in navigation, the ship often shakes violently and moves in six degrees of freedom including sway, surge, heave, roll, pitch, yaw, so there is a lot of recyclable mechanical energy around the ship. In these six degrees of freedom motions, due to the small damping of ship rolling motion, the rolling motion of the ship is the most violent, especially in medium and large merchant ships, according to Ref. [29]–[31], the mechanical energy contained in ship rolling is more abundant. However, at present, there is no such power generation device that uses the mechanical energy generated by the ship rolling, therefore, if we can develop a device to use the mechanical energy of ship rolling, it will have greater recycling and utilization value.

In order to prove the rationality of the above device for indirect utilization of wave energy, the Newmark- method is used as the basis of numerical simulation. In this article, the Newmark-β method is used for numerical analysis. The relevant materials about the Newmark-β method are as follows: Wang et al. [32] investigated numerically the effects of Triangle Groove Strips on Vortex-induced Vibration suppression of marine riser. Ulveseter et al. [33] mentioned that the equation of motion was solved by the time integration scheme Newmark-β method. According to Ref. [34], Behrooz Farshi, Abbas Assadi mentioned that the Newmark-β method for step by step integration was used to find the dynamic response of the structure. Besides, Wang et al. [35] mentioned that the effect of the grooved elastic damping component on the friction-induced vibration was investigated by using both experimental and numerical analysis. Therefore, the Newmark-β method is a numerical analysis method commonly used for the dynamic response of structures, the method has good accuracy and stability. The literature mentioned above has solved relevant problems by using the Newmark-β method.

In a word, it has practical application value to recover the mechanical energy produced by the medium and large merchant ship rolling, so a device fixed on the deck of a ship is proposed. This energy recovery device converts mechanical energy into electric energy, so it is a power generation device that indirectly utilizes wave energy. The device is fixed on the ship deck, and the ship plays the role of converting wave energy into mechanical energy. When the ship is sailing, the device can realize the conversion from wave energy to mechanical energy and then to electrical energy, and there is no need for large energy storage equipment, it can directly provide some electricity for ship electronic equipment, cabin lighting, etc. So as to reduce the energy consumption of the ship. Therefore, this kind of energy recovery device using ship rolling has better practical significance.
II. DESIGN SCHEME

A. OVERALL DESIGN

Generally, the merchant ship will have six degrees of freedom motion on the voyage, among which the rolling amplitude is the largest. When the ship is sailing in different sea conditions, the rolling angle of the ship is about $5 \sim 10^\circ$ in the condition of small wind and waves, and the rolling angle of the ship is about $20 \sim 30^\circ$ in the condition of big wind and waves. Fig. 1 is the schematic diagram of the rolling motion of the ship. Based on this situation, an energy recovery device is designed to recover the mechanical energy contained in the ship during rolling caused by wave motion.

The design drawing of the device is shown in Fig. 2. The structure of the device mainly includes the frame of the device, slider, track, spring, spring rod, pulley, belt, gearbox and electromagnetic generator (generator). The main parameters are shown in Table 1.

The energy recovery device uses the mechanical energy generated by the rolling motion of the ship in navigation, converting mechanical energy into electrical energy. The rolling motion of the ship is mainly caused by wave motion, so the device is a kind of power generation device that uses wave energy indirectly. The device is installed in pairs on the port and starboard sides of the deck and the purpose is to reduce the impact on ship’s stability, and the plane of the frame must be perpendicular to the middle line plane of the ship. All devices shall be installed as close to the side of the ship as possible. The frame of the device is fixed to the deck, the installation method is shown in Fig. 3.

The ship equipped with energy recovery device rolls at sea under the influence of wind and waves, besides, part of the mechanical energy contained in the ship is transferred to the

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**FIGURE 1.** Schematic diagram of rolling motion of the ship.

**FIGURE 2.** Energy recovery device and its partial enlarged drawing.

**TABLE 1.** Device dimensions.

| Description | Value |
|-------------|-------|
| $r$         | 8 m   |
| $L$         | 9.5 m |
| $B$         | 1.2 m |
| $H$         | 5 m   |
slider in the device. Specifically, suppose the ship rolls to the left, so the device will roll to same direction. Because the slider is on a curved surface, the slider with wheels will be affected by the gravity component and move to the left along the arc track. Then, the belt fixed on the slider is pulled to move along the frame of the device, next, because the length of the belt is invariable, the belt movement causes the generator pulley to spin, so as to realize the generator produces electricity. Similarly, it is the same law when the ship rolls to the right.

B. DETERMINING SHIP
It is better to choose the medium type ship with a general load to install the device. The installation of the device needs more deck space. The tanker is the ideal one for the device thanks to having less deck machinery. The selected tanker is called “Lian Huan Hu”, owned by COSCO SHIPPING, the world’s largest shipping companies. The main parameters of the ship are shown in Table 2.

| Description | Value  |
|-------------|--------|
| Length      | 183 m  |
| Beam        | 32 m   |
| Draft       | 11.4 m |
| Tonnage     | 50000 ton |

III. MATHEMATICAL MODEL
A. MOTION ANALYSIS
The energy recovery effect of the device has an important relationship with the parameters of ship rolling motion and the relevant parameters of the device. The main relevant parameters include rolling angle of the ship, rolling period of the ship, track radius, quality of the slider, system effective damping coefficient $C$ (effective damping coefficient), spring group elasticity coefficient $K$ (elasticity coefficient) and so on.

Considering the influence on the device of the rolling angle and rolling period of the ship, firstly, the mathematical model equation of the rolling period of the ship and rolling angle of the ship is established. Then, the moving mode of the slider along the track is analyzed. The motion equation of the slider in the device installed on the deck is established and the energy recovery effect and utilization effect of the device is analyzed by combining the rolling of the ship and the motion of the slider.

1) ANALYSIS OF SHIP ROLLING MOTION
The rolling angle and rolling period of ship are two main variables of describing the severity of ship. It is found that the motion of the slider in the device is affected by the rolling angle velocity of the ship $\omega_1$ and rolling angle of the ship $\theta_1$ according to actual survey, so the equation of ship rolling motion is established.

It is considered that ship rolling motion is simple harmonic motion, the equations are as follows:

$$\omega_1 = asin(bt + \frac{\pi}{2})$$ (1)
$$\theta_1 = \int_0^t \omega_1 du$$ (2)

If equation (1) is carried into equation (2)

$$\theta_1 = \int_0^t asin(bu + \frac{\pi}{2})du$$ (3)

where $\omega_1$ is rolling angular velocity of the ship, $\theta_1$ is the rolling angle of the ship, $a$ and $b$ are parameters for controlling ship rolling angle and period.

2) MOTION ANALYSIS OF SLIDER IN THE DEVICE
The energy recovery device is installed on the ship deck. When the ship is rolling the slider will incline, at this time, the slider on the circular track will produce a downward gravity component. This gravity component is the direct force driving the slider movement, the model of slider on the tangent line of track (spring not considered) is established, as shown in Fig. 4. In addition, when analyzing the motion characteristics of the slider, the slider should be considered as a mass point, besides, the position of the mass point is the axis of the spring rod.
The gravity component of the slider in motion is \( F(t) \), according to Newton’s second law:

\[
F(t) = mg \sin(\theta_1 + \theta_2) = m\ddot{x}
\]  

(4)

The movement of the slider on the arc track is decomposed into small elements, the tangent line of track is a straight line, the same time, the movement of the slider on the unit length track is regarded as a straight line. Considering the function of spring (the spring installation method is shown in Fig. 2), the mathematical model of slider vibration along the slope is established. The mass-spring-damper oscillator model is considered. Therefore, the motion equation of the slider is given as follows: The equation in tangent direction with one degree of freedom is modeled by a second-order linear equation, thus:

\[
m\ddot{x} + C\dot{x} + Kx = F_x(t)
\]  

(5)

Also, equation \( F_x(t) \) and \( \theta_2 \):

\[
mg \sin(\theta_1 + \theta_2) = F_x(t)
\]

(6)

\[
\theta_2 = x/r
\]

(7)

Bring equation (3), (6) and (7) into equation (5):

\[
m\ddot{x} + C\dot{x} + Kx = mg \sin \left( \int_0^t \sin(bu + \frac{\pi}{2})du + \frac{x}{r} \right)
\]

(8)

where \( m \) is slider quality, \( C \) is effective damping coefficient, \( K \) is elasticity coefficient, \( x \) is linear displacement of the slider along the track, \( \dot{x} \) is linear velocity of the slider along the track, \( \ddot{x} \) is linear acceleration of the slider along the track, \( r \) is radius of circular track, \( g \) is gravitational acceleration, \( \theta_1 \) is rolling angle of the ship, \( \theta_2 \) is angular displacement of the slider.

B. NEWMARK-\( \beta \) METHOD

The Newmark-\( \beta \) method is an explicit integral solution to the time differential equation and the mainstream method of discrete finite element equations [36], [37]. In fact, the method modifies the assumption of linear acceleration. There are two parameters \( \gamma \) and \( \beta \) being introduced into the velocity and displacement expression at the moment of \( t + \Delta t \). When \( \gamma \) and \( \beta \) take certain reasonable values (\( \gamma \geq 0.5, \beta \geq \gamma/2 \)), the calculation process is considered to be unconditionally convergent. The accuracy of the Newmark-\( \beta \) method depends on the size of the time step. The error can be considered as minimal when the value of time step is small enough, and it has effective calculation accuracy. The Newmark-\( \beta \) can be used to solve the vibration equation of the slider, basic input conditions include the slider quality, the effective damping coefficient, the excitation \( F_x(t) \), then the motion parameters of the slider at the next moment can be obtained. In addition, other methods are compared and found that the Newmark-\( \beta \) method has a faster calculation speed and able to meet the requirements of calculation accuracy by adjusting the time step.

Firstly, according to the Newmark-\( \beta \) method the motion displacement, \( u \), velocity, \( \dot{u} \), and acceleration, \( \ddot{u} \), can be formulated respectively as:

\[
\begin{align*}
 u_{t+\Delta t} &= (u_t + \dot{u}_t \Delta t) + \left( \frac{1}{2} - \beta \right) \ddot{u}_t + \beta \ddot{u}_{t+\Delta t} \Delta t^2 \\
 \dot{u}_{t+\Delta t} &= \frac{1}{\beta \Delta t^2} (u_{t+\Delta t} - u_t) - \left(1 - \frac{\gamma}{\beta}\right) \dot{u}_t + \frac{1}{\beta \Delta t} \ddot{u}_t - \left(\frac{1}{2\beta} - 1\right) \dddot{u}_t \\
 \ddot{u}_{t+\Delta t} &= \frac{1}{\beta \Delta t^2} (u_{t+\Delta t} - u_t) - \frac{1}{\beta \Delta t} \ddot{u}_t - \left(\frac{1}{2\beta} - 1\right) \dddot{u}_t
\end{align*}
\]

(9) - (11)

where the subscript \( t \) or \( t + \Delta t \) denotes the time level. The parameters, \( \gamma \) and \( \beta \) are two adjustable weighting constants. According to the studies in Refs. [37], [38], the method is unconditionally stable for \( \beta \geq \gamma/2 \) and \( \gamma \geq 0.5 \). The scheme is second-order accurate only for \( \gamma = 0.5 \). For all other values of \( \gamma \), the method is only first-order accurate. So, in order to achieve second-order accuracy, the choice of values for \( \gamma \) is strictly limited to a single value 0.5. Setting \( \beta \) to various values between 0 and 0.5 can give a wide range of results. Typically \( \beta = 1/4 \), which yields the constant average acceleration method is used. Therefore, the values of \( \gamma \) and \( \beta \) are 0.5 and 0.25 in Newmark-\( \beta \) method, which gave unconditionally stable and second-order accurate for non-dissipative linear systems.

Considering equation (5) at time \( t + \Delta t \), we have:

\[
m\ddot{x}_{t+\Delta t} + C\dot{x}_{t+\Delta t} + Kx_{t+\Delta t} = F_x(t_{t+\Delta t})
\]

(12)

Since only \( \ddot{x}_{t+\Delta t}, \dot{x}_{t+\Delta t} \) and \( u_{t+\Delta t} \) are unknown, by combining equations (10) - (12), they can be obtained at time \( t + \Delta t \). Finally, equation (12) can be further simplified:

\[
[\dddot{u}_{t+\Delta t}] = [\dddot{x}_{t+\Delta t}] = [\tilde{F}]
\]

(13)

where

\[
[\dddot{u}_{t+\Delta t}] = [K] + \frac{1}{\beta \Delta t^2} m + \frac{\gamma}{\beta \Delta t} C
\]

\[
[F] = F_x(t_{t+\Delta t}) + \left[ \frac{1}{\beta \Delta t^2} u_t + \frac{1}{\beta \Delta t} \dot{u}_t + \left( \frac{1}{2\beta} - 1 \right) \ddot{u}_t \right] m
\]

\[
+ \left[ \frac{\gamma}{\beta \Delta t} u_t + \left( \frac{\gamma}{2\beta} - 1 \right) \dot{u}_t + \left( \frac{\gamma - 1}{2\beta} - 1 \right) \ddot{u}_t \right] C
\]

(14) - (15)
The initial values \( u_0, \dot{u}_0 \) and \( \ddot{u}_0 \) are given, then, using the Newmark-\( \beta \) method, we can get the results of \( t + \Delta t \) time, including linear acceleration, linear velocity, linear displacement of the slider, Fig. 5 is the calculation flow chart. Linear displacement and linear velocity of the slider provide the basis for the later calculation of device parameters, including power of device, the angular displacement of the slider \( \theta_2 \) and linear velocity of the slider.

**C. CALCULATION OF POWER**

According to the above calculation, the corresponding linear velocity of the slider \( \dot{x} \) in a certain period is obtained, then \( P = Cx^2 \) is used to calculate the average power of the device, the relevant equations are as follows:

\[
p_{av} = \frac{1}{T} \int_0^T F \dot{x} \, dt \tag{16}
\]

Take equation (5) into equation (16):

\[
p_{av} = \frac{1}{T} \int_0^T [(m\ddot{x} + C\dot{x} + Kx)\dot{x}] \, dt \tag{17}
\]

where \( P_{av} \) is average power, \( T \) is time.

Considering that the slider will return to its initial position (the middle position) in a complete cycle, according to the law of conservation of mechanical energy, the mechanical energy of the slider at the beginning and the end of a cycle is the same, the \( \dddot{x} \) is 0 at this moment, so the \( \Delta m\ddot{x} \) is 0; similarly, the length of the spring does not change, \( \Delta Kx \) is 0. Equation (17) can be further simplified. The simplification approach refers to Ref. [39]–[41]. thus:

\[
p_{av} = \frac{1}{T} \int_0^T C \dot{x}^2 \, dt \tag{18}
\]
In addition, consider the loss of energy in the process of transmission in the device, the transmission efficiency $\eta$ is 80% by refer to general mechanical transmission losses, so the transmission energy loss of the device is 20%.

$$p_{av} = \frac{1}{T} \int_0^T C \dot{x}^2 dt \quad (19)$$

Through the establishment of the above mathematical model, the relevant parameters are found out, which will provide an algorithm model for MATLAB simulation.

IV. SIMULATION

A. DETERMINE SIMULATION SCHEME

In order to simulate and calculate the power generation effect of the device fixed on the ship, it is necessary to select the type and working conditions of the simulated ship (typical ship). The ship has been selected above, and its working conditions are as follows:

According to the actual investigation of ship technicians who work on the tanker "Lian Huan Hu", the maximum rolling angle of the ship is at least $5 \sim 10^{\circ}$ in windy and wave days, and $10 \sim 20^{\circ}$ in severe sea conditions, the rolling period of the ship is generally $10 \sim 13$ s. In order to make the research more universal, the rolling angle of $6^{\circ}$ is chosen as the simulation condition.

The above ship type and the structure of the energy recovery device have been determined. In addition, the slider quality is one of the important factors affecting the power of the device, and power increases with slider quality. In order to facilitate quantitative analysis, the quality of the selected slider is constant, the quality is 200 kg. Therefore, the following analysis mainly considers the influence of the effective damping coefficient and elasticity coefficient on the generating power of the device.

In order to evaluate the energy recovery power of the device, using MATLAB software to simulate. According to the Fig. 5 shows that Newmark-$\beta$ method, relevant mathematical model and calculation steps, the program is programmed. Then the program is input to MATLAB software for numerical simulation and the simulation results are got.

B. PRELIMINARY SIMULATION

According to the above argument, parameters are determined, which include the track radius is 8 m, the rolling angle of the ship is $6^{\circ}$, the rolling period of the ship is 10.5 s, quality of the slider is 200 kg.

In addition, due to the influence of the time step of Newmark-$\beta$ method on the computational accuracy, the reasonable value of time step needs to be determined. As can be seen from Table 3, the influence of the time step on the calculation accuracy is illustrated by comparing the average amplitude of the slider’s angular displacement (shown in Fig. 6 (b)) corresponding to different time steps. When the value of the adjacent time step is different, the average amplitude is slightly different. The difference is lowered when the time step decreases. When the time steps are 0.03 s, 0.02 s and 0.01 s, the differences between the average amplitude values of the three-time steps’ calculation results are less than 0.019, and calculation rate is considered. Therefore, taking the time step as 0.02 s can meet the accuracy requirement.

There are only two input variables in MATLAB simulation system, including effective damping coefficient $C$ and elasticity coefficient $K$. Now taking a group of numerical inputs randomly from these two variables for simulation calculation, $K = 350$ N/m and $C = 100$ N/(m/s) are chosen, then, the introduction of natural frequency $\omega_n$ and damping ratio $\epsilon$ can make the calculation more general. In this way, the results become more general as they are made independent of the mass involved, there $\omega_n = 1.323$ Hz, $\epsilon = 0.189$. The various rules of the gravity component, the angular displacement, the linear velocity of the slider and the instantaneous power of the device with time are obtained, which are shown in Fig. 6. In order to display the change rules and accuracy of each parameter better in the figure, the images of six cycles (about 63 s) of motion are analyzed and taken.

Fig. 6(a) is an image shows the variation of the gravity component of the slider with time in the device. The curve in the figure is periodic and close to a sine curve. After the second period, the curve tended to be stable in the figure. It illustrates that the motivation of slide motion always exists, the peak value of the gravity component is 708.6 N.

It can be seen in Fig. 6(b) that the slider moves along the track when the ship is rolling, the angle displacement ($\theta_2$) curve of the slider is approximate sinusoidal, the maximum angle of slider unilateral movement is $17.94^{\circ}$.

The linear velocity of the slider on the track changes periodically, the linear velocity is relatively small in the first cycle, next, the curve is approximate sinusoidal, and the maximum linear velocity of the slider is 1.5 m/s in Fig. 6(c).

According to Fig. 6(d), the instantaneous power value of the device is small in the first cycle. And in the third period, the average power curve tends to be stable. Therefore, it can be seen the device is working continuously. In this state, the maximum instantaneous power of the device is 223.5 W.

In Fig. 6, the curve in each figure changes regularly with time, and the values of simulation curve are convergent, therefore, the movement of the energy recovery device is relatively stable, it indicates that the device can generate

**TABLE 3. The time step of Newmark-$\beta$ method.**

| Time step (s) | 0.005 | 0.01 | 0.02 | 0.03 | 0.04 | 0.05 | 0.06 | 0.08 | 0.10 |
|---------------|-------|------|------|------|------|------|------|------|------|
| Average amplitude (°) | 17.358 | 17.350 | 17.332 | 17.313 | 17.289 | 17.262 | 17.230 | 17.193 | 17.116 |
| Difference (°) | 0.008 | 0.018 | 0.019 | 0.024 | 0.027 | 0.032 | 0.037 | 0.077 |
electricity continuously under the ideal stable simulation condition.

C. AVERAGE POWER ANALYSIS OF THE DEVICE

According to the analysis in B. preliminary simulation, the rolling device of the ship can work continuously. However, the damping ratio $\varepsilon$ and natural frequency $\omega_n$ of the system are arbitrarily given, the instantaneous power and average power calculated by simulation may not be the optimal values (the maximum value satisfying certain conditions), considering different combinations of $\varepsilon$ and $\omega_n$ aims to find as much power as possible. In the following, several typical combinations in the simulation experiment are extracted for comparative analysis. In addition, the maximum angular displacement of the slider should be considered comprehensively to avoid excessive angles.

The $\omega_n$ is 1 Hz in Fig. 7(a), the $\varepsilon$ is independent variable. The data of average power of device, maximum angular displacement of slider and gravity component of slider (When the curve changes steadily, the peak gravity component of each combination of $\varepsilon$ and $\omega_n$) are obtained by simulation. Then taking $\varepsilon$ as the independent variable, the average power, the maximum angular displacement and the gravity component as the dependent variables, the figures are drawn. Similarly, when $\omega_n$ is 1.118, 1.225, 1.323, 1.414 and 1.5 Hz, the results are also obtained by simulation and the curves are drawn. As shown in Fig. 7.

1) An image with $\omega_n$ is 1 Hz in Fig. 7(a). When $\varepsilon = 0.075$, it is not hard to learn the average power reaches maximum, and the maximum average power is 89.90 W. At this time, the instantaneous power curve of the device and the angular displacement curve of the slider (these two kinds of curves are not provided in the paper due to limited space) are irregular. From the angular displacement curve of the slider, it can be seen that the value of the angular displacement is always positive, which means the slider has been sliding on the left or right side of the track. And it is found that the angular displacement is too large. The instantaneous power curve of the device is very irregular and the value of the instantaneous power is small, so the combination of $\varepsilon$ and $\omega_n$ is not desirable.

2) As shown in Fig. 7(b) an image with $\omega_n$ is 1.118 Hz. According to the analysis chart, when $\varepsilon = 0.2012$, the average power reaches the maximum, it is 80.70 W. The maximum average power is smaller than 89.90 W above.

3) Fig. 7(c) is an image with $\omega_n$ is 1.225 Hz. According to the analysis chart, when $\varepsilon = 0.0612$, the average power reaches the maximum, and the maximum average power is 174.11 W.

4) It can be seen from Fig. 7(d) an image with $\omega_n$ is 1.323 Hz. When $\varepsilon = 0.1512$, the average power reaches the maximum, and the maximum average power is 107.35 W.
5) An image with $\omega_n$ is 1.414 Hz is given in Fig. 7(e). When $\varepsilon$ is 0.2298, the average power reaches the maximum, and the maximum average power is 62.74 W.

6) As shown in Fig. 7(f), it is an image with $\omega_n = 1.5$ Hz. When $\varepsilon$ is 0.3333, the average power reaches the maximum, and the maximum average power is 42.05 W. The average power curve of the slider has a little change and the average power value is small.

In Fig. 7, the gravity component curve of the slider changes in the same direction as the maximum angular displacement curve of the slider. Except for Fig. 7(a), the maximum angular displacement of other images is not too large, the angular displacement curves are regular and reasonable. Also, there is no direct relationship between the gravity component and the main research (average power). Here the gravity component is only for reference.

In order to facilitate the analysis, the collection of average power curves of the energy recovery device is given when $\omega_n$ is 1, 1.118, 1.225, 1.323, 1.414 Hz respectively, as shown in Fig. 8(a), the Fig. 8(a) is modified comparison figure.
the value range of $\varepsilon$ is $0 \sim 0.38$. It is important to point out that the abscissa of curve $\omega_n = 1.5$Hz is the upper axis of Fig. 8(a), the value range of $\varepsilon$ is $0.10 \sim 0.45$.

Next, the maximum average power of the energy recovery device is given when $\omega_n$ is 1, 1.118, 1.225, 1.323, 1.414, 1.5 Hz respectively, as shown in Fig. 8(b).

As can be seen from the comparison of the images in Fig. 8(a) and Fig. 8(b), compared with other groups of data, when $\omega_n$ is 1.225 Hz, the average power of the device is larger which is 174.11 W. However, the distance between the six groups of $\omega_n$ numerical conditions in Fig. 7 is about 0.1. It is conjectured that there may be a power value greater than 174.11 W ($\omega_n = 1.225$ Hz) between $\omega_n = 1.118$ and $\omega_n = 1.323$. So, adding $\omega_n = 1.173$ between 1.118 and 1.225, adding $\omega_n = 1.275$ between 1.225 and 1.323 to simulate analysis, as shown in Fig. 9. The purpose is to obtain a better value by comparing the maximum average power values corresponding to $\omega_n = 1.173$, $\omega_n = 1.225$, and $\omega_n = 1.275$.

In Fig. 9(a), when $\omega_n$ is 1.173 Hz, $\varepsilon$ is 0.1279, the average power reaches the maximum, it is 113.7 W. In Fig. 9(b), when $\omega_n$ is 1.275 Hz, $\varepsilon$ is 0.0981, the average power reaches the maximum, it is 146 W. And their maximum average power are still less than 174.11 W. Therefore, it can be proved that $\omega_n = 1.225$ Hz, $\varepsilon = 0.0612$ is the relative optimal combination. Next, input the optimal combination into MATLAB for simulation calculation. The curves of the gravity component, the angular displacement, the linear velocity of the slider and the instantaneous power of the device are obtained. As shown in Fig. 10.

According to Fig. 10(a), the gravity component of the slider in the device reaches the maximum in the third cycle, the maximum value is 1390 N, the curve after the third cycle is approximate sinusoidal. Fig. 10(b) shows that the angular displacement of the slider on the device increases gradually in the first two cycles (about 1 $\sim$ 20 s), at this time, the slider is continuous climbing stage along the track, after the third period, it was stable, and the maximum angular displacement of the slider is 44.71°. Fig. 10(c) shows that the movement of the slider is reciprocating, after the third period, the linear velocity curve tended to be stable, and the curve was approximate sinusoidal. Fig. 10(d) shows that the instantaneous peak power of the device is very small at first, it is 50 W, instantaneous power increases gradually during 0 $\sim$ 30 s, the maximum instantaneous power is 392.8 W when $t$ is 35 s. After the second period, the motion of the slider tended to be stable according to Fig. 10(b) and Fig. 10(c), it proves that the device operates continuously.
When $\omega_n = 1.225$ Hz and $\varepsilon = 0.0612$, the combination of $\omega_n = 1.225$ Hz and $\varepsilon = 0.0612$ is feasible. Combining the calculation formula of natural frequency and damping ratio, the following formula is obtained: When $\omega_n = 1.225$, $\varepsilon = 0.0612$, the corresponding elasticity coefficient $K = 300$ N/m and effective damping coefficient $C = 30$ N/(m/s). At this time, the relevant parameters of the energy recovery device are shown in Table 4.

**TABLE 4. Relevant parameters of energy recovery device.**

| Symbol | Description                  | Value     |
|--------|------------------------------|-----------|
| $r$    | Track radius                 | 8 m       |
| $m$    | Quality of the slider        | 200 kg    |
| $K$    | Elasticity coefficient       | 300 N/m   |
| $C$    | Effective damping coefficient of the system | 30 N/(m/s) |
| $\omega_n$ | Natural frequency | 1.225 Hz  |
| $\varepsilon$ | Damping ratio     | 0.0612    |
| $\Theta_1$ | The rolling angle of the ship | 6°       |
| $\Theta_2$ | The rolling period of the ship | 10.5 s   |
| $P_{av}$ | Peak power                  | 392.5 W   |
| $P_{av}$ | Average power               | 174.11 W  |

In the above analysis, the optimal parameters of the device have been given under certain conditions that rolling angle is 6° and rolling period is 10.5 s. The rolling angle and rolling period of the ship will be different in different sea areas. Theoretically, the optimal effective damping coefficient $C$ and elasticity coefficient $K$ corresponding to different sea conditions are not unique. However, once the device is installed and fixed, it is not convenient to directly change the structure or material of the device with the change of rolling angle and rolling period. Therefore, the device in Table 4 is still used as the simulation device for comparative analysis. As shown in Fig. 11, when the ship rolling angle is 5°, 6°, 7°, 8°, and the corresponding rolling period is 9.8 s, 10.5 s, 11 s, and 11.9 s respectively, the device parameters are compared. This figure shows the power curves of the device and the relevant parameters of the slider when the rolling period and rolling angle of the ship change (the Fig. 11 is a synthetic figure).

As shown in Fig. 11(a), it is a comparison figure of the gravity component of the slider. It can be seen from the figure that in several different states of rolling angle, the curves are close to a kind of sine curve. When the rolling angle is 5°, the peak value of the gravity component is 827.5 N; when the rolling angle is 6°, the peak value of the gravity component is 1393 N; when the rolling angle is 7°, the peak value is 1543 N; when the rolling angle is 8°, the peak value is 1576 N. In addition, it can be seen from the simulation that when the rolling angle of the ship is 5°-8°, the gravity component of the slider is increases with the rolling angle of the ship.

In Fig. 11(b), it is a comparison figure of the angular displacement of the slider. It can be seen from the figure that the curves are similar to a kind of sine curve, indicating that the slider moves back and forth along the track. When the ship
FIGURE 11. Comparison figure of simulation results.
rolling angle is 5°, the peak value of the angle displacement is 27.7°; when the ship rolling angle is 6°, 7° and 8° respectively, the peak value of the angle displacement is about 45°.

As shown in Fig. 11(c), it is the linear velocity comparison figure of the slider, and the linear velocity curves are close to a kind of sine curve. When the rolling angle increases from 5° to 8°, the peak value of the linear velocity does not increase. When the rolling angle is 6° and 7°, the linear velocity curves are relatively high, about 3.65 m/s.

Fig. 11(d) is the power comparison figure of the device. It can be seen from the figure that the instantaneous power always exists under different rolling angles and rolling periods. When the ship rolling angle is 5°, the instantaneous power peak value is 230.3 W, and the average power is 115.92 W; the instantaneous power peak value is 392.8 W and the average power is 174.11 W when the rolling angle is 6°; when the rolling angle is 7°, the instantaneous power peak of the device is 399 W and the average power is 197.5 W; the instantaneous power peak is 307 W and the average power is 166.9 W when the rolling angle is 8°. From these four cases, the average power of the device is the highest when the rolling angle is 7°, which is 197.5 W. The reason is that: not only the rolling angle of the ship affects the power of the device, but also the rolling period affects the power. When the angle is increased, the average power does not always increase, so the power and rolling angle may not be a single linear relationship.

By analyzing the curves in Fig. 11, it can be seen that the power value generated by the same device is different and the slider motion is also different when the ship is sailing in different sea conditions (the rolling angle and rolling period of the ship are different).

D. SIMULATION OF COMPLICATED WORK CONDITION

The above analysis mainly shows the feasibility of the device and simulates the response of the device under a single sea condition (an ideal stable condition in which the environment does not change over time). The actual sea conditions are more complex. Due to the limitation of experimental conditions, it is impossible to measure the rolling motion of this typical ship (50000 t). Besides, this device is suitable for most of ships and is not just for the selection of typical ships. Therefore, the angular velocity data of a scientific research ship rolling recorded during operation in Bohai Bay of China is measured. In the paper, the local waves in a certain period of winter in Bohai Bay of China are taken as the simulation conditions, and the continuous 240 s in a certain period of time is selected for analysis. The ship’s rolling conditions under complex sea conditions are given, and the response of the device is discussed further. The rolling angular velocity of the ship is shown in Fig. 12.

Then, the mathematical model of the device is combined with the angular velocity data of the scientific research ship, and the relevant parameters of the device are obtained through MATLAB simulation, these relevant parameters are as follows:

The curve of power, gravity component, angular displacement and linear velocity is given by simulated, as shown in Fig. 13. For the convenience of comparative analysis, the device structure parameters are still referred to Table 4. As shown in Fig. 13(a), the gravity component curve of the slider is given. The gravity component curve is nonlinear, and the peak value of gravity component is 1949 N. Fig. 13(b) and Fig. 13(c) are the angular displacement curve and angular velocity curve of the slider respectively. It can be seen from the two figures that the slider moves along the circular track with variable speed and reciprocation as the ship continues to roll. Fig. 13(d) is the instantaneous power curve of the device. It can be seen from the figure that when the ship is sailing in complex sea conditions, the power curve is fluctuant obviously with the ship rolling motion, indicating that the instantaneous power is uneven, but it can output continuously. The average power of the device is 157.5 W in 240 s under this complex sea condition by numerical analysis. Comparing the power with the power (174.11 W) of a single sea condition (rolling angle is 6°, period is 10.5 s), the average power value
FIGURE 13. Variational relationship of simulation results under complex sea conditions.
of the complex sea condition is slightly smaller, but the value deviation is not large, which shows that the device is feasible in complex sea conditions.

V. CONCLUSION

It is found that ships are prone to roll under the influence of wind and waves at sea through practical survey, and there is a large amount of recoverable mechanical energy in their rolling motion. Therefore, an energy recovery device is proposed and designed, which is fixed on the ship deck, and indirect recovery of wave energy is realized by rolling of the slider, and wave energy is converted into electric energy, which can provide part of energy for medium and large merchant ships in navigation. Taking the common medium-sized commercial ship “Lian Huan Hu” as the typical ship, according to the characteristics of its voyage, the overall design of the energy recovery device is carried out, the main parameters of the device installed on the ship deck are given in detail, and its principle is described. The 3D model of the device is built by SolidWorks, and then the model of the slider moving along the track is built. The flow chart of solving the vibration equation of the slider system is given with Newmark-β method, and the mathematical equation of the slider motion in the device is calculated and solved with the numerical analysis method.

Using MATLAB to simulate and analyze the factors that affect the power generation performance of the device, and the main research results are as follows:

1) Under a kind of stable single sea condition, the slider in the energy recovery device can realize reciprocating motion when the ship continues to roll. The device can continuously respond to the ship’s rolling motion, and can reliably provide energy for the ship.

2) In the case of the natural frequency $\omega_n = 1, 1.118, 1.225, 1.323, 1.414, 1.5$ Hz, their average power of the device are also different with the change of the damping ratio $\varepsilon$ of the system. Its trend of curve can be approximated as the quadratic curve with the opening downward, that is to say, the corresponding optimal power can always be found in the case of different natural frequency.

3) Under the condition of fixed ship’s rolling period is 10.5 s and ship’s rolling angle is 6°, amounts of data are simulated and compared, the analysis results show that the maximum average power of the device occurs at $\omega_n = 1.225$ Hz, $\varepsilon = 0.0612$, and the average power value is 174.11 W.

4) When the rolling angle and rolling period is varying, the power generated by the device and the motion rule of the slider will vary under the structure of the device is immobile. Furthermore, the power generated by the same device and the motion of the same slider will vary under different sea conditions. It is proved that the working efficiency of the device is affected by both the rolling period and the rolling angle of the ship.

5) In the complex sea conditions, the slider moves unevenly and it moves back and forth with variable speed along the track. The output power of the device is also uneven, but it can still generate continuous power on the whole, which shows that the device can recover and utilize energy under complex sea conditions.

The research in this article will provide theoretical basis and design basis for the later application of the energy recovery device in engineering, that is, the recovery and utilization of the mechanical energy generated by the rolling motion of the ship will be realized through the installation of the device. In addition, when the rolling angle of ship is $5^\circ, 6^\circ, 7^\circ, 8^\circ$ respectively, the power generated by the device and the movement of the slider are compared and analyzed through the simulation experiment. Based on the above research, it is found that there are four main factors affecting the power generation of the energy recovery device, two of these are related to the device itself: the elasticity coefficient $K$ and effective damping coefficient $C$ in the device; the others are external factors: the rolling period and rolling angle of the ship. In the future, more actual working conditions need to be considered for more complex research and real ship experiment.

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