Introduction

More than 70% of the total global international trade volume is achieved by marine transportation [1]. Under the harsh sea conditions, ships might frequently be in distress incidents which can result in great loss of the crew and the property [2]. The most effective rescuing approach at sea is the three-dimensional (3D) Air-Sea Search and Rescue which is the cooperation between the rescue ship and the helicopter. However, due to the small limited capacity of the fuel tank, the helicopter must return to the mother rescue ship to refuel when performing long-term rescue operations. When the ship is violently swaying, the helicopter cannot land on the ship, hence it should hover above the ship and refuel. In the helicopter hovering refuelling system, an emergency breakaway device needs to be installed in the refuelling pipeline. When the tension on the pipe reaches a certain tensile force value, the device can immediately disconnect the pipeline to ensure the safety of the helicopter. As a safety component, the emergency breakaway device is mainly applied industry, and there are fewer emergency breakaway devices dedicated to helicopter hovering refuelling. For instance, in 1972, Badger [3] designed a safety device for fluid pipelines. The device connects the two components through the braking bolts. When the pipeline is subjected to a certain tensile force, the two components are separated. In 1974, Strizk [4] designed a safety disconnect connector. The device was not used for pipeline oil transfer, but it was the prototype of the emergency breakaway device. In 1998, Wang and co-workers designed an emergency breakaway device for pipeline sealed by poppet valves. After the device was disconnected, both sides of the pipeline were sealed by poppet valves. On this basis, an emergency breakaway device with ball valve structure was designed in 2000. However, manual control was needed for those devices [5–7]. In 2002, Wang and Wu [8] designed a helicopter hovering refuelling system and designed a rolling ball type emergency breakaway structure. When the pipeline is subjected to tension, the ball compresses the spring by squeezing the cover to achieve valve body separation. From 2004 to 2007, Liu and Ji [9] designed a series of revolting emergency breakaway structures, which can be controlled by manual and automatic methods [10,11]. In 2005, Nimberger [12] designed a spring-ball-type emergency breakaway device. In 2017, Zhang and Zang [13] designed an emergency breakaway device for those helicopters, which were used for aircraft engine gas transfer, and calculated the strength, flow and braking force. Currently, the manufactures who produce emergency breakaway devices mainly include German RS [14], American EATON [15], US OPW [16] etc. and many companies also have researched the rapid breakaway device applied to the port oil delivery arm [17–19]. However, the existing emergency breakaway devices have a large braking force and are not suitable for helicopter hovering refuelling system. In this paper, a claw type emergency breakaway device is designed which can be reused and has a braking force of 1700–2000 N.

Design of the emergency breakaway device

As shown in Fig. 1, the breakaway structure of the claw type emergency breakaway device basically comprises an upper valve, a lower valve, a claw, a torsion spring and a pivot. The upper valve body is pressed on the lower valve body by the claws force, and the working principle is the lever principle that the pivots are the centre of rotation, and the two ends of the claws bear the force from the upper valve body and the torsion spring, respectively. The sealing structure mainly includes a poppet valve, a cross clamp type and an O-ring. There are two requirements for the design of the sealing structure: during the refuelling of the helicopter, there should be no leakage between the upper and lower valve bodies; after the breakaway structure is disconnected, the pipeline can be shut down. The design of the sealing poppet valve should also meet the flow requirements of the device. According to the Helicopter Ship System Specification, the velocity of oil in helicopter pipeline should be <4.5 m/s, and the helicopter refuelling flow rate is 215–250 L/min. Thus, the maximum diameter of the poppet valve and the inner diameter of the valve body can be calculated [20].

1 Design of the emergency breakaway device

The helicopter hovering refuelling emergency breakaway device consists of two parts: the breakaway structure and the sealing structure. The breakaway structure is to quickly separate the helicopter from the refuelling ship when the pipeline is subjected to a certain tension (1700–2000 N) during the helicopter hovering refuelling; the sealing structure shut down the pipeline after the breakaway device is separated. According to the requirements of the emergency breakaway device, a claw-type emergency breakaway device is designed.

2.1 Structural design and working principle

As shown in Fig. 1, the breakaway structure of the claw type emergency breakaway device basically comprises an upper valve, a lower valve, a claw, a torsion spring and a pivot. The upper valve body is pressed on the lower valve body by the claws force, and the working principle is the lever principle that the pivots are the centre of rotation, and the two ends of the claws bear the force from the upper valve body and the torsion spring, respectively. The sealing structure mainly includes a poppet valve, a cross clamp type and an O-ring. There are two requirements for the design of the sealing structure: during the refuelling of the helicopter, there should be no leakage between the upper and lower valve bodies; after the breakaway structure is disconnected, the pipeline can be shut down. The design of the sealing poppet valve should also meet the flow requirements of the device. According to the Helicopter Ship System Specification, the velocity of oil in helicopter pipeline should be <4.5 m/s, and the helicopter refuelling flow rate is 215–250 L/min. Thus, the maximum diameter of the poppet valve and the inner diameter of the valve body can be calculated [20].
Breakaway state: When the tension of the oil pipeline reaches the braking force of the emergency breakaway device, as shown in Fig. 2, the upper valve body is run away from the claw to realise the separation of the upper and lower valve bodies, and the upper and lower poppet valves are pressed against the valve seat under the action of spring force to achieve the sealing function.

2.2 Parameter design of the emergency breakaway device

The key part of the claw type emergency breakaway device is the claws that decide the braking force, and the impact factors of the braking force are obtained by analysing the force of the claw. As shown in Fig. 3, \( c_1 \) is the action point of the force from the upper valve body to the claw, \( F_1 \) is the positive pressure of the contact surface between the claw and the upper valve body, \( F_1 \) is the friction force between the upper valve body and the claw; \( c_2 \) is the action point of the torsion spring and the force is \( F_2 \).

Force analysis of the claws can list the moment balance equation with the fulcrum O of the claw as the centre of rotation

\[
F_1 \cos \alpha x_1 + F_1 \sin \alpha y_1 + F_1 \sin \alpha z_1 = F_1 \cos \alpha y_1 + F_2 (x_2 + y_2^2)
\]

where \( \alpha \) is the claw contact angle.

The torsion spring stiffness is influenced by the material and size, so it can be used as a variable in the design. Since the torsion spring stiffness is influenced by the material and size, it is difficult to install when the pre-tightening angle is too large, so the initial selection is 4° and 8° for convenient installation. Since the torsion spring stiffness is influenced by the material and size, the selection range is wide, so it can be used as a variable in the design.

\[
F_1 = M(\theta + \phi)/\sqrt{x_2^2 + y_2^2}
\]

where \( M \) is the moment of the pre-tightening angle and its value is 0.12.

The tension mainly used to overcome the restraint of the claws. In addition, the factors such as system self-weight have little influence on the tension, so they are neglected. Then the tension \( F \) is as follows:

\[
F = n(F \cos \alpha + F_2 \sin \alpha)
\]

where \( n \) is the number of claws.

The tensile equation of the emergency breakaway device can be derived from above (1)–(4)

\[
F = n(F \cos \alpha + F_2 \sin \alpha)
\]

The position of the rotation centre of the claw O \((x_0, y_0)\) determines the values of \( x_{c1}, y_{c1}, x_{c2} \) and \( y_{c2} \) in (1). When the structure and dimensions of the emergency breakaway device are determined, the number of the claws \( n \) and the position of the centre of rotation centre O can be determined, so they can be regarded as the fixed parameters to calculate the remaining parameters. When the emergency breakaway device is pulled off, the rotation angle \( \phi \) of the claw is a fixed value. It can be concluded from (5) that the main impact factors of the braking force are claw contact angle \( \alpha \), pre-tightening angle \( \theta \) and spring stiffness \( M \). In order to ensure that the upper and lower valve bodies are tightly connected in the initial position of the breakaway process, and if the contact angle \( \alpha \) is too large, the braking force is insufficient. So, the initial selection is 15°, 25° and 35°. According to the structure and size of the claw, it is difficult to install when the pre-tightening angle is too large, so the initial selection is 4° and 8° for convenient installation. Since the torsion spring stiffness is influenced by the material and size, the selection range is wide, so it can be used as a variable in the design.
### 2.3 Design parameter optimisation

In order to select the parameters of the impact factors that meet the requirements of the braking force, the ADAMS is used to simulate the dynamics of the preselected model. As shown in Fig. 4, 3D model is created in SolidWorks based on the pre-selected parameters. After the model is imported into ADAMS, motion pairs and forces are added between parts. The lower valve body is fixed and the upper valve body is added a vertical upward velocity. Then the force of the upper valve body which is simulated and analysed output is the force value of the emergency breakaway device in the process of sports.

The design requirement of braking force for the emergency breakaway device is 1700–20 000 N. The stiffness of torsion spring is 47, 77 and 119 N mm/°, according to (5) and spring diameter standard. According to the parameters of pre-selected impact factors, a three-factor and three-level orthogonal tests were conducted. The simulation was performed by Adams and the results were analysed by the range analysis. The results are shown in Table 1.

#### Table 1 Orthogonal test

| Number | Claw contact angle (α) | Pre-tightening angle (θ) | Spring stiffness (M) | Breaking force (F), N |
|--------|------------------------|--------------------------|----------------------|----------------------|
| 1      | 15                     | 0                        | 47                   | 720                  |
| 2      | 15                     | 4                        | 77                   | 1407                 |
| 3      | 15                     | 8                        | 119                  | 2163                 |
| 4      | 25                     | 0                        | 77                   | 1134                 |
| 5      | 25                     | 4                        | 119                  | 1546                 |
| 6      | 25                     | 8                        | 47                   | 640                  |
| 7      | 35                     | 0                        | 119                  | 1678                 |
| 8      | 35                     | 4                        | 47                   | 689                  |
| 9      | 35                     | 8                        | 77                   | 1165                 |
| k1     | 1430                   | 1177                     | 683                  | —                    |
| k2     | 1106                   | 1214                     | 1235                 | —                    |
| k3     | 1177                   | 1322                     | 1795                 | —                    |
| R      | 323                    | 145                      | 1112                 | —                    |

#### Table 2 Orthogonal experimental linear regression analysis results

| Modal                      | Unstandardised coefficient B | Significance |
|----------------------------|------------------------------|--------------|
| (constant)                 | 240.405                      | 0.384        |
| claw contact angle         | −12.633                      | 0.131        |
| pre-tightening angle       | 18.167                       | 0.347        |
| torsion spring stiffness   | 15.318                       | 0.001        |

The size of the range shows that the order of the influence of each factor on the braking force is torsion spring stiffness $M$, claw contact angle $α$, pre-tightening angle $θ$. Linear regression analysis was performed on the orthogonal test data by statistical product and service solutions (SPSS), and the results are shown in Table 2. According to the results in Table 2, the linear regression equation can be listed as

$$F = -12.633α + 18.167θ + 15.318M + 240.405$$

According to the linear regression equation, the values of the pre-selected parameters of the braking force can be calculated, the results are shown in Table 3. According to the design requirements of the braking force (1700–2000 N), the selected numbers are 3 ($α=15°$, $θ=4°$, $M=119$ N mm/°), 9 ($α=25°$, $θ=8°$, $M=119$ N mm/°), 12 ($α=25°$, $θ=8°$, $M=119$ N mm/°), and these parameters are used in the experiment.

### 3 Verification experiment

The hydraulic circuit diagram used in the experiment is shown in Fig. 5. According to the hydraulic circuit diagram, the test bench is built, as shown in Fig. 6, it includes water tanks, water pumps, throttle valves, relief valves, electromagnetic flow meters, electromagnetic pressure gauges, grippers etc. The flow rate and pressure are measured by an electromagnetic flowmeter (XY-LDE-40, Jiangsu Xiyuan Instrument Technology Co., Ltd) and an electromagnetic pressure gauge (XY-BST8800, Jiangsu Xiyuan Instrument Technology Co., Ltd).

The testing machine is used to simulate the working state of the helicopter hovering refuelling, and the experimental results are analysed and summarised. Then the emergency breakaway device that meets the braking force requirement can be selected. The testing machine model is CTM8010 shown in Fig. 7. The emergency breakaway device is mounted on the testing machine through a special gripper, and the braking force of the breakaway device is measured at a constant velocity.

The emergency breakaway device not only meets the requirements of the braking force but also ensures its good sealing capacity. Due to limited laboratory conditions and the density of kerosene is close to water, water instead of kerosene is used as the flow medium to verify the sealing capacity of the breakaway device.
During the experiment, the emergency breakaway device is connected to the position of the experimental system 6 shown in Fig. 5, and used as a special gripper to connect with the machine. The tension value during the experiment is measured by the load cell. In the water-passing experiment, the throttle valves 3, 9 and the relief valve 10 are regulated to adjust the system flow and pressure to the helicopter refuelling requirements. Industrial computer and data acquisition card were used to collect signals of the electromagnetic flowmeter and electromagnetic pressure gauge. Those signals are processed by Matlab to obtain the pressure and flow curves of the system.

4 Experimental results

According to the design results, processing experiments were carried out on the parameters of Table 3 with serial numbers of 3, 9, 12 and 18. According to the design results, the processing experiments were carried out with the parameters of serial numbers 3, 9, 12 and 18 in Table 3. The contact angles of the claws are 15°, 25°, 35°, the pre-tightening angles are 4°, 8° and the spring stiffness is 119 N mm°. Table 4 shows the average value of the braking force of the five repeated tensile tests with different parameters.

It can be seen from Table 4 that only the average braking force of the emergency breakaway device with parameter No. 3 meets the design requirements (1700–2000 N), so the experimental analysis is carried out by selecting the pre-tightening angle of 8°, the claw contact angle of 25°, and the spring stiffness of 119 N mm°.

The braking force and flow pressure curve of the emergency breakaway device are measured by the water passing experiment, as shown in Fig. 8. It can be seen that the braking force of the emergency breakaway structure is within the required range (1700–2000 N). The distance of the curve from point A to point B is about 5.5 mm, which is the pre-tightening process of the testing machine, and the emergency breakaway device is not subject to tension. The distance of the curve from point B to point C is about 4 mm. During this process, the force is increasing until it is out of the limit position. The emergency breakaway device is completely broke away from point C to point D, and the force of the claw on the upper valve body sharply decreases to zero. The tension fluctuates from point D to point E. According to the experimental phenomena, this is the case that five claws in the claw-type emergency breakaway device cannot be disengaged at the same time. From the flow curve analysis, there is no obvious leakage when the emergency breakaway device is disconnected. During the gradual separation of the breakaway device, the poppet valve gradually closes the valve port, and the flow rate is reduced. When the breakaway structure is completely disengaged, the valve port is shut down and the flow rate is reduced to zero.

During the test, when the breakaway structure is completely detached, the emergency breakaway device still has a little volume of water splashing out. Through the analysis of the structure of the device, it is found that the inevitable leakage caused by the

### Table 3 Calculation results of braking force

| Number | Claw contact angle | Pre-tightening angle | Spring stiffness | Breaking force |
|--------|-------------------|---------------------|-----------------|----------------|
| 1      | 15                | 4                   | 47              | 843.524        |
| 2      | 15                | 4                   | 77              | 1303.064       |
| 3      | 15                | 4                   | 119             | 1946.42        |
| 4      | 15                | 8                   | 47              | 916.192        |
| 5      | 15                | 8                   | 77              | 1375.732       |
| 6      | 15                | 8                   | 119             | 2019.088       |
| 7      | 25                | 4                   | 47              | 717.194        |
| 8      | 25                | 4                   | 77              | 1176.734       |
| 9      | 25                | 4                   | 119             | 1820.09        |
| 10     | 25                | 8                   | 47              | 789.862        |
| 11     | 25                | 8                   | 77              | 1249.402       |
| 12     | 25                | 8                   | 119             | 1892.758       |
| 13     | 35                | 4                   | 47              | 590.864        |
| 14     | 35                | 4                   | 77              | 1050.404       |
| 15     | 35                | 4                   | 119             | 1693.76        |
| 16     | 35                | 8                   | 47              | 663.532        |
| 17     | 35                | 8                   | 77              | 1123.072       |
| 18     | 35                | 8                   | 119             | 1766.428       |
structure of the device itself. The filling part in Fig. 9 is the leakage amount of the device itself. Then the leak collection box is installed on the testing machine to measure the leakage of the emergency breakaway device. Table 5 shows the amount of self-leakage of the emergency breakaway device at different pressures and different extend speeds.

It can be seen from Table 5 that the leakage of the device is related to the pressure and the extended speed. Leakage increases with the increase of pressure and decreases with the increase of extended speed. However, the leakage of the device is within an acceptable range. During the helicopter hovering refuelling process, when an emergency situation occurs, the upper and lower valve bodies will be quickly disengaged, and the leakage of the device should be less than the value in Table 5.

5 Conclusions

According to the design requirements, the breakaway structure and sealing structure of the claw type emergency breakaway device is designed. Through the analysis, the dimensions of each part of the device are calculated, and the equation for calculating the braking force is obtained. Four sets of parameters that meet the design requirements of the braking force are obtained. According to the simulation results, the prototype of the emergency breakaway device was processed and the tensile test was carried out. With the tensile test, the structural parameters that meet the requirements are selected, and the breakaway device is designed to meet the requirements of the braking force (1700–2000 N). The parameters are the torsion spring pre-tightening angle of 8°, the claw contact angle of 25°, and the spring stiffness 119 N·mm/°. The water-passing experiment was carried out on the emergency breakaway device, which verified that the emergency breakaway device has a good sealing ability.

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7 References

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Table 4 Breaking force with different parameters

| Number | Claw contact angle (α), ° | Pre-tightening angle (θ), ° | Average braking force, N |
|--------|---------------------------|-----------------------------|--------------------------|
| 1      | 15                        | 4                           | 2564                     |
| 2      | 25                        | 4                           | 1460                     |
| 3      | 25                        | 8                           | 1826                     |
| 4      | 35                        | 8                           | 1466                     |

Table 5 Leakage rate of claw emergency breakaway device

| Number | Pressures, MPa | Extend speeds, mm/min | Leakage, ml |
|--------|----------------|-----------------------|-------------|
| 1      | 0.20           | 50                    | 108         |
| 2      | 0.20           | 200                   | 71          |
| 3      | 0.30           | 50                    | 127         |
| 4      | 0.30           | 200                   | 87          |
| 5      | 0.40           | 50                    | 176         |
| 6      | 0.40           | 200                   | 135         |

Fig. 7 Tenile testing device

Fig. 8 Displacement-flow–tension curve of claw emergency breakaway device

Fig. 9 Self-leakage of emergency breakaway device
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