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Design and Performance Analysis for the Low-Power Holding Mechanism of the All-Electric Subsea Gate Valve Actuator

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Abstract: The all-electric subsea gate valve actuator is one of the critical components of the all-electric subsea production control system. To bridge the gap of the low-power holding mechanism in the all-electric subsea gate valve actuator of the subsea production system, minimize the power consumption and cable number for control and improve the open-position keeping performance of all-electric subsea gate valve actuator, this paper proposed a novel low-power holding mechanism for the all-electric subsea gate valve actuator which can be applied to all-electric subsea gate valve actuators with various valve sizes and process pressure ratings. The proposed low-power holding mechanism uses an electromagnet as a driving element, combines the spiral transmission and the cam-like transmission, and only requires a holding force of approximately 2–7% of the maximum load of the closing spring to keep the valve open. The proposed low-power holding mechanism converts the axial force of the closing spring into the circumferential force, which substantially reduces the output force required for the driving element of the low-power holding mechanism and the number of the actuator's control cables. Analytic models are created for the lockable maximum load of the closing spring and the permissible stroke of the locking tab with regard to the design variables. The parameter effects and the corresponding sensitivities are discussed by numerical analysis. The design parameters and the lockable maximum load of the closing spring of the low-power holding mechanism are obtained.

Keywords: all-electric subsea production control system; all-electric subsea gate valve actuator; low-power holding mechanism; configuration research; performance analysis

1. Introduction

Late in the last century, in order to meet the needs of deep-water gas and oil exploitation, subsea equipment suppliers began to develop and improve the all-electric subsea control technology [1]. The all-electric Subsea Production Control System (AE), a leading-edge and breakthrough technology, which didn't need hydraulic fluid and was driven completely by electric power, was proposed and developed [2]. The key benefits of the AE system compared to the conventional Multiple Electric-Hydraulic Control System (MEH) are as follows [3–10]: the simplified system structure, enhanced control performance, improved system reliability and availability, increased the oil recovery factor and environmental friendliness. Furthermore, the advantages of AE system also include: the decreased power consumption [11,12], reduced Capital expenditure(CAPEX) and Operating

expenditure(OPEX) [13–16], strengthened intelligence [17,18]. Therefore, with the improvement of key techniques of the AE system, the AE system will replace the MEH system as a trend in the future [19–21].

The all-electric subsea gate valve actuator is one of the critical components of the AE system [6,7], which is used to open and close the production gate valve for the subsea Xmas tree. Its performance directly affects the reliability and safety of the AE system. Similar to the traditional hydraulic-driven subsea gate valve actuator, in the opening and closing processes of the valve, the all-electric subsea gate valve actuator is required to have a failsafe close ability, Remotely Operated Vehicle(ROV) override function, subsea ambient pressure compensation and valve position indication [22,23]. The all-electric subsea gate valve actuator uses the motor as the actuation unit, and the service life cycle of the motor is lower than that of the hydraulic cylinder. Therefore, to ensure the high reliability of the actuator, the actuation device should be designed with regard to redundancy [24]. In addition, the subsea gate valve is a normally open valve. When the valve is opened, the power system needs to resist the resilience of the failsafe closing mechanism, which consumes a lot of energy in long term operation [25]. Thus, to reduce the energy consumption, the low-power holding mechanism is required for the all-electric subsea gate valve actuator to ensure that the valve remains open with small power consumption.

The research and development of all-electric subsea gate valve actuators are in the early phase. A few oil companies like Cameron and FMC have carried out relevant research on it. Cameron's all-electric subsea gate valve actuator realized the opening and closing operation of the subsea gate valve through a high-power "drive" motor, and was equipped with a low-power "clutch" motor to ensure that the valve remained open by means of a friction-based mechanism. Further, a closing spring was used as the failsafe close solution [6,7]. FMC's all-electric subsea gate valve actuator powered the built-in motion control system via locally stored power from rechargeable batteries and capacitors to open and close the subsea gate valve and was equipped with the ROV interface. No springs were required to move a valve to a failsafe position [8,11]. At present, the all-electric subsea gate valves actuator of each oil company are still in the initial application stage, and the key technology of has not yet matured. Winther-Larsen [26] developed a functional design specification for the all-electric subsea gate valve actuator and proposed a design for an all-electric subsea gate valve actuator, as well as the motor control system layout based on the specification. Jon Berven [27] discussed failsafe solutions of the all-electric subsea gate valve actuator and created the calculation method of the peak load and the applied torque of the actuator. Wang [24] proposed the multi-motor parallel redundant method to drive the valve and designed the structure of an all-electric subsea valve actuator. Xiao [28] developed a subsea all-electric Christmas tree gate valve actuator and analyzed the sealing process of the gate valve. Liu [25] proposed a new pressure compensation all-electric control gate valve and actuator integrated structure, which substantially reduced the high-pressure hydraulic resistance during valve closure.

According to the above analysis, it is obvious that some designs and researches have been done for the all-electric subsea gate valve actuator. However, most of the existing researches on the all-electric subsea gate valve actuator do not concentrate on the low-power holding mechanism, which was the unique composition of the all-electric subsea gate valve actuator. Therefore, a low-power holding mechanism of the all-electric subsea gate valve actuator is proposed in this paper, which uses an electromagnet as a driving element and combines spiral transmission and cam-like transmission, the corresponding mechanical and kinematic models are also established, respectively. The analytic expression of the relation is derived, respectively, for the lockable maximum load of the closing spring and the permissible stroke of the locking tab with regard to the design variables. The parameter effects and the corresponding sensitivities are analyzed as well. The lockable maximum load of the closing spring and the design parameters of the low-power holding mechanism are obtained.

This present work is aimed at developing a novel and feasible low-power holding mechanism of the all-electric subsea gate valve actuator, which can minimize the power consumption and cable

number for control and improve the open-position keeping performance of an all-electric subsea gate valve actuator, only requiring a holding force of approximately 2%~7% of the maximum load of the closing spring to keep the valve open and with the ability to be applied to all-electric subsea gate valve actuators with various valve sizes and process pressure ratings, to bridge the gap of the low-power holding mechanism in the all-electric subsea gate valve actuator of the subsea production system.

The rest of the paper is organized as follows: Section 2 briefly introduces the working process of the all-electric subsea gate valve actuator proposed, followed by a full presentation of the configuration, work principle and advantage of the proposed failsafe closing and low-power holding mechanism. Section 3 presents the mechanical and kinematic models. In Section 4, the detailed discussion of the parameter effects and sensitivity analysis is performed based on the established model and results. Finally, conclusions are given in Section 5.

2. Research on the Configuration of Failsafe Closing and Low-Power Holding Mechanism

Based on the function of the all-electric subsea gate valve actuator, a configuration of the all-electric subsea gate valve actuator is proposed in this paper, as shown in Figure 1, wherein the structure mainly includes the subsea gate valve assembly, failsafe closing and low-power holding mechanism, subsea ambient pressure compensating mechanism, screw drive mechanism, multi-motor redundant driving mechanism, valve position indicating mechanism and ROV interface. During the valve opening process, multi-motor redundant driving mechanism or ROV drives the valve stem and gate downward to open the subsea gate valve through the screw drive mechanism. After the subsea gate valve is fully opened, the low-power holding mechanism locks the failsafe closing mechanism to ensure that the valve remains open with the minimum power consumption. When the all-electric subsea production control system breaks down, the low-power holding mechanism and multi-motor redundant driving mechanism are power-off, the failsafe closing mechanism drives the valve stem and gate upward to realize the emergency failure close.

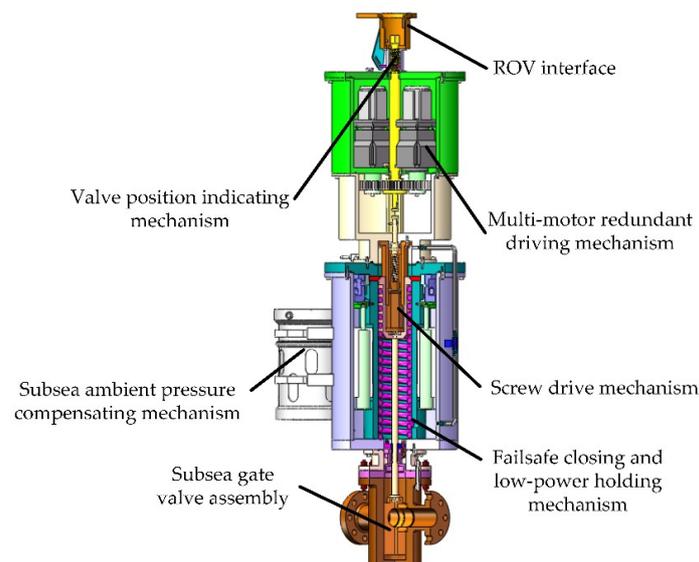


Figure 1. Schematic diagram of the all-electric subsea gate valve actuator.

Failsafe closing mechanisms of the subsea valve include the mechanical failsafe closing mechanism and Uninterrupted Power System (UPS) failsafe closing mechanism [29,30]. As the service life cycle of the subsea Xmas tree or manifold is usually around 20 years, long term continuous power supply makes the UPS failsafe closing mechanism unpractical. The mechanical failsafe closing mechanism is most commonly used for this purpose. In the application of the mechanical failsafe closing mechanism for a subsea valve actuator, the spring-type configuration is reliable and widely used [4,25]. However, the force of the closing spring needs to be big enough to oppose the high-pressure fluid in the valve

when a failure close occurs. So when the valve is opened, the power system needs to resist the large spring forces, which consume a lot of energy [25]. Thus, the low-power holding mechanism also needed to be designed in the actuator of this paper. In addition, a single Electric Subsea Control Module (ESCM) is claimed to control up to 32 functional groups [6]. However, a single ESCM can control up to 16 functional groups including sensors in an engineering application [12], which puts forward requirements for the number of the actuator's control cables.

Based on the above analysis, the low-power holding mechanism is proposed in this paper, as shown in Figure 2, in which a spiral groove is machined inside the locking ring sleeve; a thread and four cam-like notches are machined outside the locking ring and inside the bottom, respectively. Four locking tabs are adopted due to the higher symmetry and stability of this configuration, compared with the design of two locking tabs. Furthermore, the requirements of this configuration for the size of the locking ring are moderate, because each locking tab needs a cam-like notch of the locking ring. The higher the number of locking tabs, the higher the number of cam-like notches necessary, which leads the size of the locking ring being easily unable to meet the requirements. Thus, the configuration of four locking tabs has advantages in realizing product function, improving product performance and meeting design constraints.

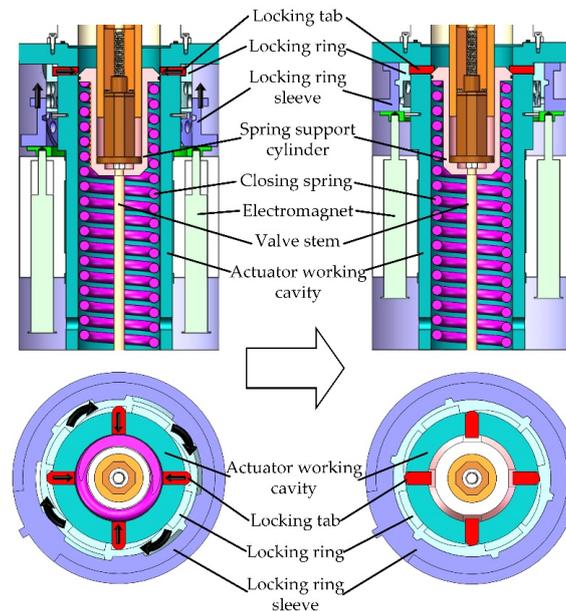


Figure 2. The locking and holding process of the low-power holding mechanism.

The locking and holding process of the low-power holding mechanism is shown in Figure 2. The driving mechanism pushes the spring support cylinder against the closing spring until the valve is fully opened, at which point the closing spring is at the limit, and the electromagnets are energized and push the locking ring sleeve upward. The locking ring sleeve drives the locking ring to rotate forward through the spiral groove. Meanwhile, the locking ring drives locking tabs to move inward through the cam-like notch. The locking process of the low-power holding mechanism is completed when the locking tabs fully enter the actuator working cavity and make contact with the slope of the spring support cylinder to block the cylinder moving back. After the locking process, the holding process of the low-power holding mechanism is started. In the holding process, the locking tab and the spring support cylinder constrain each other, and the position of the locking tab is restricted by the locking ring, to ensure that the valve remains open with the minimum power consumption.

When a valve fail-close occurs, the electromagnets are de-energized, thereby the position of the locking tab is not restricted by the locking ring. The unlocking process is opposite to the locking process, and the closing spring force is the driving force for the unlocking process. The closing spring drives the

locking tabs to move outward through the slope of the spring support cylinder. Meanwhile, the locking tabs drive the locking ring to rotate backward through the cam-like notch surface, and the locking ring pushes the locking ring sleeve downward through the spiral groove. The unlocking process is completed when the locking tab and the spring support cylinder are separated. After the unlocking process, the closing spring continues to release rapidly and drives the spring support cylinder and the valve stem upward until the valve is fully closed.

The proposed low-power holding mechanism converts the axial force of the closing spring into the circumferential force by the spiral groove and the cam-like notches, which substantially reduces the output force required for the driving element of the low-power holding mechanism and increases the feasibility and reliability of the holding mechanism. In addition, different from the form of Cameron's "clutch" motor, the proposed low-power holding mechanism is driven by the electromagnet, which effectively reduces the number of the actuator's control cables.

When a fail-close occurs, the maximum load produced by the high-pressure fluid which the all-electric subsea gate valve actuator can oppose depends on the force of the closing spring. According to the configuration of the low-power holding mechanism, the maximum load of the closing spring is related to the size of the low-power holding mechanism. However, the permissible stroke of the locking tab also depends on the size of the low-power holding mechanism and needs to be larger than the given working stroke, which will form a constraint for the optimization of the locking and holding ability of the low-power holding mechanism. Thus, it is necessary to analyze the mechanical and kinematic models of the low-power holding mechanism in detail.

3. Mechanical Model and Kinematic Analysis of the Low-Power Holding Mechanism

3.1. Mechanical Model of the Low-Power Holding Mechanism

The low-power holding mechanism adopts four locking tabs to share the maximum load of the closing spring. Figure 3 shows the force analysis of the spring support cylinder and single locking tab in the holding process, in which $F_{m1}/4$ is the maximum load of the closing spring shared by the single locking tab, R_{16} and R_{61} are the interactive force between the spring support cylinder and the single locking tab, R is the radial component force acting on the spring support cylinder by the single locking tab in the symmetrical direction, R_{63} and R_{36} are the interactive force between the single locking tab and the actuator working cavity, F_r and F'_r are the interactive radial thrust between the single locking tab and the locking ring.

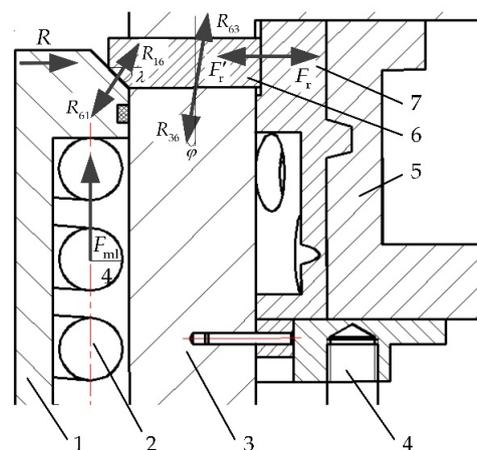


Figure 3. The force analysis of the spring support cylinder and single locking tab in the holding process. 1—Spring support cylinder, 2—Closing spring, 3—Actuator working cavity, 4—Electromagnet, 5—Locking ring sleeve, 6—Locking tab, 7—Locking ring.

In the holding process, the spring support cylinder and the single locking tab are in a state of mechanical equilibrium. Figure 4 shows the force diagram of the spring support cylinder and the single locking table as shown in Figure 4a, the relation between R_{61} and F_{m1} can be expressed as:

$$R_{61} = \frac{F_{m1}}{4 \cos(\lambda - \varphi)}, \tag{1}$$

where λ is the angle of the slope of the single locking tab; φ is the friction angle; F_{m1} is the maximum load of the closing spring.

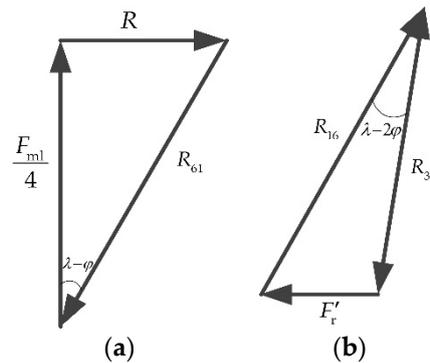


Figure 4. The force diagram of (a) the spring support cylinder and (b) the single locking tab.

As shown in Figure 4b, the relation between R_{61} and F_r can be given as:

$$\frac{R_{16}}{\sin(90^\circ + \varphi)} = \frac{R_{61}}{\sin(90^\circ + \varphi)} = \frac{F_r'}{\sin(\lambda - 2\varphi)} = \frac{F_r}{\sin(\lambda - 2\varphi)}, \tag{2}$$

Combining Equations (1) and (2), the relation between F_r and F_{m1} can be expressed as:

$$F_r = \frac{R_{16} \sin(\lambda - 2\varphi)}{\sin(90^\circ + \varphi)} = \frac{F_{m1} \sin(\lambda - 2\varphi)}{4 \cos(\lambda - \varphi) \cos \varphi}, \tag{3}$$

Figure 5 shows the force diagram of the contact between the single locking tab and the locking ring in the holding process, in which R_{67} and R_{76} are the interactive forces between the single locking tab and the locking ring. To facilitate the analysis, the force diagram in Figure 5 is simplified, as shown in Figure 6. This assumes that O_1 is the center of the actuator working cavity; O_2 is the center of the notch of the locking ring; O_3 is the center of the curved surface of the locking tab; C is the contact point between the locking tab and locking ring; the straight line, MN, is the normal direction of the contact point C; S is the starting point of the notch of the locking ring, which is the contact point between the notch and the actuator working cavity; E is the ending point of the notch of the locking ring, which is the intersection of the notch and the line O_1O_2 ; δ is the angle of the notch of the locking ring. The angle between R_{76} and MN is φ since R_{76} is the resultant force of normal pressure and friction; the direction of F_r is parallel to O_1O_3 . Because the notch of the locking ring and the actuator working cavity are not concentric, a circumferential force F_t is introduced at the contact point C, which is not only the resistance to overcome by the low-power retaining mechanism in the holding process, but also the driving force of the backward rotation of the locking ring in the unlocking process. The locking ring rotates around the center O_1 in the locking or unlocking process, so F_t is perpendicular to the segment O_1C . Draw line segments begin in the direction of F_r and R_{76} , and end at points A and B, respectively, where the segment $AB \parallel F_t$ and passes through the center O_1 .

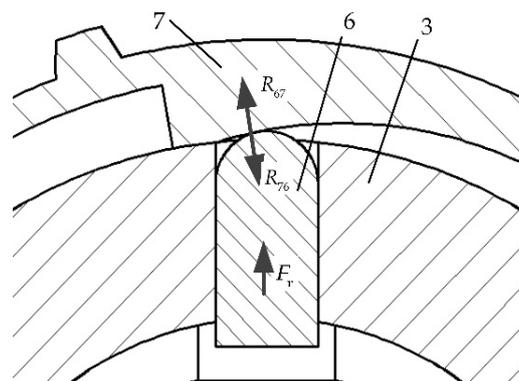


Figure 5. The force diagram of the contact between the single locking tab and the locking ring in the holding process. 3—Actuator working cavity, 6—Locking tab, 7—Locking ring.

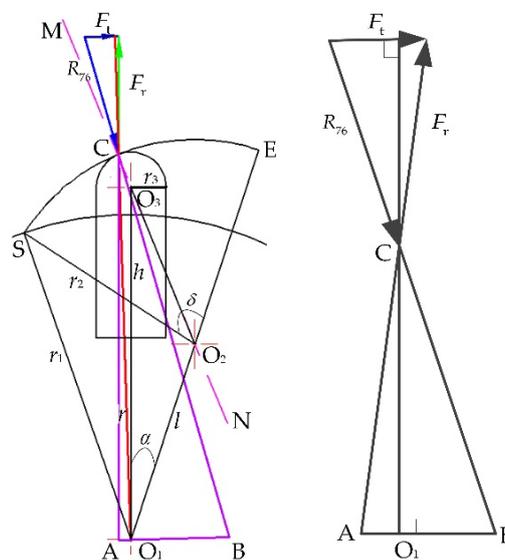


Figure 6. The force analysis of the contact between the single locking tab and the locking ring in the holding process.

As shown in Figure 6, the relationship between F_t and F_r can be obtained by calculating the relationship between AB and AC. The relevant design parameters of the low-power holding mechanism are listed in Table 1, and the intermediate parameters in the calculation process are listed in Table 2.

Table 1. The relevant design parameters of the low-power holding mechanism.

Parameter Description	Symbol
Radius of the actuator working cavity	r_1
Radius of the notch of the locking ring	r_2
Radius of the locking tab	r_3
Center distance between the actuator working cavity and the locking ring notch ($ O_1O_2 $)	l
Radial deflection angle of the center of the locking ring notch ($\angle O_2O_1O_3$)	α

Table 2. The intermediate parameters in the calculation process.

Parameter Description	Symbol
Center distance between the actuator working cavity and the locking tab ($ O_1O_3 $)	h
Distance between the center of the actuator working cavity and the contact point C ($ O_1C $)	r

In $\triangle O_1O_2O_3$ and $\triangle O_1O_3C$, h, r and $\angle CO_1O_3$ can be obtained, respectively:

$$h^2 = (r_2 - r_3)^2 + l^2 + 2l(r_2 - r_3) \cos \left[\alpha + \arcsin \left(\frac{l \sin \alpha}{r_2 - r_3} \right) \right], \tag{4}$$

$$r^2 = r_3^2 + h^2 + 2r_3h \cos \left[\arcsin \left(\frac{l \sin \alpha}{r_2 - r_3} \right) \right] \text{ and} \tag{5}$$

$$\angle CO_1O_3 = \arccos \left(\frac{r^2 + h^2 - r_3^2}{2rh} \right). \tag{6}$$

As $F_r \parallel O_1O_3$ and $CO_1 \perp AB$, $\angle CAB$ can be given as:

$$\angle CAB = \frac{\pi}{2} - \angle CO_1O_3 = \frac{\pi}{2} - \arccos \left(\frac{r^2 + h^2 - r_3^2}{2rh} \right). \tag{7}$$

In $\triangle O_1O_2C$, $\angle O_1CO_2$ can be expressed as:

$$\angle O_1CO_2 = \arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right). \tag{8}$$

And $\angle ACB = \angle O_2CA - \varphi = \angle CO_1O_3 + \angle O_1CO_2 - \varphi$, so $\angle ACB$ can be given as:

$$\angle ACB = \arccos \left(\frac{r^2 + h^2 - r_3^2}{2rh} \right) + \arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right) - \varphi. \tag{9}$$

Combining Equations (3), (7) and (9), the relation between F_t and F_{ml} can be expressed as:

$$F_t = \frac{F_r \sin \angle acb}{\sin \angle abc} = \frac{F_{ml} \sin(\lambda - 2\varphi) \sin \left[\arccos \left(\frac{r^2 + h^2 - r_3^2}{2rh} \right) + \arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right) - \varphi \right]}{4 \cos(\lambda - \varphi) \cos \varphi \cos \left[\arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right) - \varphi \right]}. \tag{10}$$

Since four locking tabs are adopted to share the maximum load of the closing spring, the total circumferential force is $4F_t$. Therefore, the total resistance torque between the locking ring and the locking ring sleeve can be expressed as:

$$T = 4F_t \cdot r. \tag{11}$$

Figure 7 shows the force analysis of the locking ring sleeve and the locking ring in the holding process. Assuming that the driving force of the locking ring sleeve is fully sustained by the spiral groove of the locking ring, the actuator working cavity applies no force on the locking ring sleeve. In the holding process, the locking ring tends to rotate backward to drive the locking ring sleeve to move upward. In Figure 7, F_D is the output force of the electromagnet, R_{85} is the resultant force of the housing acting on the locking ring sleeve, R_{75} and R_{57} are the interactive force between the locking ring sleeve and the locking ring, R_{37} is the resultant force of the actuator working cavity acting on the locking ring, β is the spiral angle of the locking ring.

As shown in Figure 7, the force to resist the backward rotation of the locking ring is the circumferential resultant force of R_{57} and R_{37} , and satisfies:

$$\frac{R_{57} \sin(\beta + 2\varphi)}{\cos \varphi} \cdot R = 4F_t \cdot r, \tag{12}$$

where R is the spiral radius of the locking ring; $\beta = \arctan \left(\frac{S}{2\pi R} \right)$, and S is the pitch.

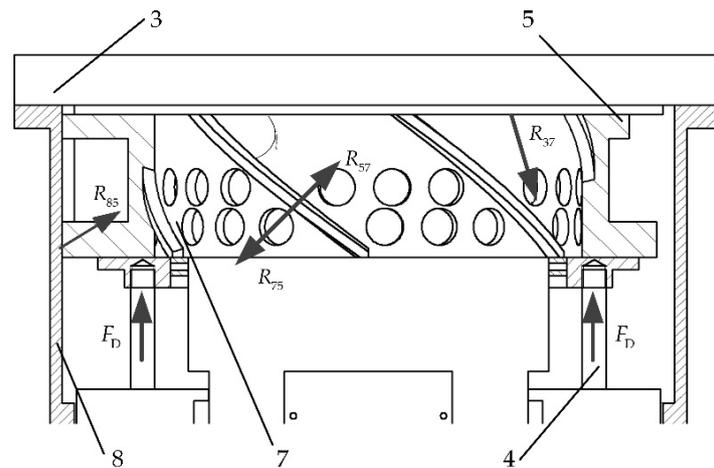


Figure 7. The force analysis of the locking ring sleeve and the locking ring in the holding process. 3—Actuator working cavity, 4—Electromagnet, 5—Locking ring sleeve, 7—Locking ring, 8—Housing.

The relation between R_{57} and F_D can be expressed as:

$$R_{57} = \frac{F_D \cos \varphi}{\cos(\beta + 2\varphi)}. \tag{13}$$

By substituting Equations (10) and (13) into Equation (12), the lockable maximum load of the closing spring can be obtained, that is:

$$F_{ml} = \frac{F_D \cos(\lambda - \varphi) \cos \varphi \cos \left[\arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right) - \varphi \right] \tan(\beta + 2\varphi) \cdot R}{\sin(\lambda - 2\varphi) \sin \left[\arccos \left(\frac{r^2 + h^2 - r_3^2}{2rh} \right) + \arccos \left(\frac{r_2^2 + r^2 - l^2}{2r_2r} \right) - \varphi \right] \cdot r}, \tag{14}$$

where

$$h = \left\{ (r_2 - r_3)^2 + l^2 + 2l(r_2 - r_3) \cos \left[\alpha + \arcsin \left(\frac{l \sin \alpha}{r_2 - r_3} \right) \right] \right\}^{\frac{1}{2}}, \tag{15}$$

$$r = \left\{ r_3^2 + h^2 + 2r_3h \cos \left[\arcsin \left(\frac{l \sin \alpha}{r_2 - r_3} \right) \right] \right\}^{\frac{1}{2}}. \tag{16}$$

3.2. Kinematic Analysis of the Low-Power Holding Mechanism

In the locking or unlocking process, the locking tab moves linearly in the cam-like notch of the locking ring, and the locking ring rotates around the actuator working cavity. Figure 8 shows the relative position between the locking tab and the locking ring in the locking state and unlocking state, respectively. Thus, the constraint condition for the locking tab to complete the locking and unlocking actions can be expressed as:

$$L \leq [L], \tag{17}$$

where $[L]$ is the permissible stroke of the locking tab; L is the working stroke of the locking tab.

The permissible stroke of the locking tab depends on the size of the low-power holding mechanism. To facilitate the analysis, the movement of the locking ring and the locking tab is equivalent so that the locking ring remains fixed, while the locking tab not only moves linearly in the notch but also rotates around the actuator working cavity during the locking and unlocking processes. Figure 9 shows the kinematic analysis of the unlocking process of the low-power holding mechanism. Assuming that O_{32} is the center of the curved surface of the locking tab in an unlocking state; O_{31} is the intersection of the segment O_1O_{32} and the arc formed by the rotation of O_3 around O_1 , m is the length of the segment

O_1O_{32} . The other symbols have the same values as depicted in Figure 6. The permissible stroke of the locking tab, $[L]$, can be expressed as:

$$[L] = m - h. \tag{18}$$

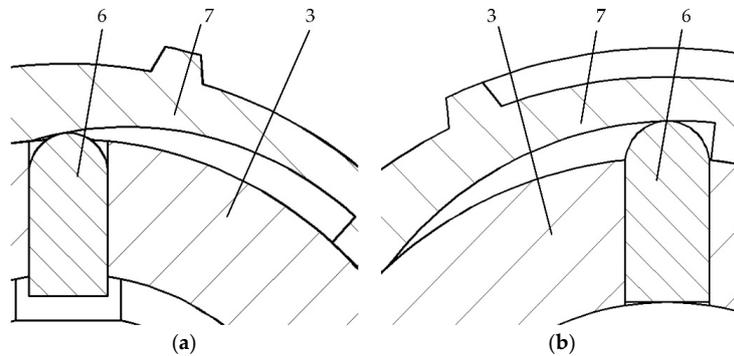


Figure 8. The relative position between the locking tab and the locking ring in (a) the locking state, (b) the unlocking state. 3—Actuator working cavity, 6—Locking tab, 7—Locking ring.

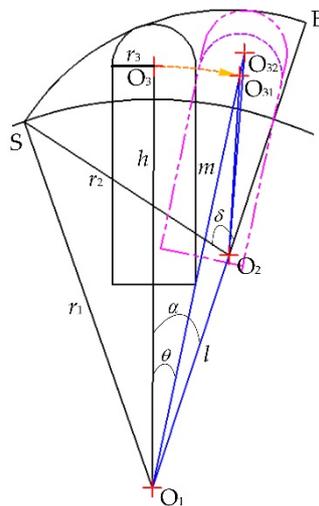


Figure 9. The kinematic analysis of the unlocking process of the low-power holding mechanism.

As shown in Figure 9, the rotation angle of the locking ring in the locking or unlocking process θ can be given as:

$$\theta = \alpha - \angle O_{32}O_1O_2 = \alpha - \arcsin\left(\frac{r_3}{r_1}\right). \tag{19}$$

So $\angle O_1O_{32}O_2$ and m can be expressed as follows, respectively:

$$\angle O_1O_{32}O_2 = \arcsin\left(\frac{l \sin \angle O_{32}O_1O_2}{r_2 - r_3}\right) = \arcsin\left(\frac{lr_3}{r_1(r_2 - r_3)}\right), \tag{20}$$

$$m^2 = (r_2 - r_3)^2 + l^2 + 2l(r_2 - r_3) \cos\left[\arcsin\left(\frac{r_3}{r_1}\right) + \arcsin\left(\frac{lr_3}{r_1(r_2 - r_3)}\right)\right]. \tag{21}$$

By substituting Equations (15) and (21) into Equation (18), the permissible stroke of the locking tab $[L]$ can be rewritten as:

$$\begin{aligned}
 [L] &= m - h \\
 &= \left\{ (r_2 - r_3)^2 + l^2 + 2l(r_2 - r_3) \cos \left[\arcsin \left(\frac{r_3}{r_1} \right) + \arcsin \left(\frac{lr_3}{r_1(r_2 - r_3)} \right) \right] \right\}^{\frac{1}{2}} - \left\{ (r_2 - r_3)^2 + l^2 + 2l(r_2 - r_3) \cos \left[\alpha + \arcsin \left(\frac{l \sin \alpha}{r_2 - r_3} \right) \right] \right\}^{\frac{1}{2}}. \quad (22)
 \end{aligned}$$

4. Numerical Analysis and Discussion

4.1. Parameter Effects

The effects of the design parameters in Table 1 on the lockable maximum load of the closing spring F_{ml} and the permissible stroke of the locking tab $[L]$, respectively, are analyzed in this section. Assume that the parameters of the low-power holding mechanism are: $\lambda = 45^\circ$, $\varphi = 5.7^\circ$, $S = 850$ mm, $R = 183$ mm, $F_D = 1600$ N, the required working stroke of the locking tab $L = 8$ mm, and the design variable ranges are listed in Table 3.

Table 3. The design variable ranges.

Design Variable Description	Variable	Variable Range
Radius of the actuator working cavity	r_1	130~150 mm
Radius of the notch of the locking ring	r_2	130~150 mm
Radius of the locking tab	r_3	10~20 mm
Center distance between the actuator working cavity and the locking ring notch	l	30~40 mm
Radial deflection angle of the center of the locking ring notch	α	35~45°

Figure 10 shows the variable effects on the lockable maximum load of the closing spring F_{ml} . F_{ml} can be increased by increasing r_2 and/or reducing α . According to Equations (5), (10) and (11), in the holding process, the resistance arm, r , increases with the increase in r_2 or decrease in α , while the resistance force, F_t , decreases with the increase in r_2 or decrease in α , and the reduction in F_t is greater than the increment in r , resulting in the decrease in the total resistance torque T . Thus, with a given maximum lockable load, F_{ml} , the required force, F_D , of the electromagnet can be reduced by increasing r_2 and/or decreasing α ; in other words, with a given output force, F_D , of the electromagnet, the maximum lockable load, F_{ml} , can be increased by increasing r_2 and/or decreasing α .

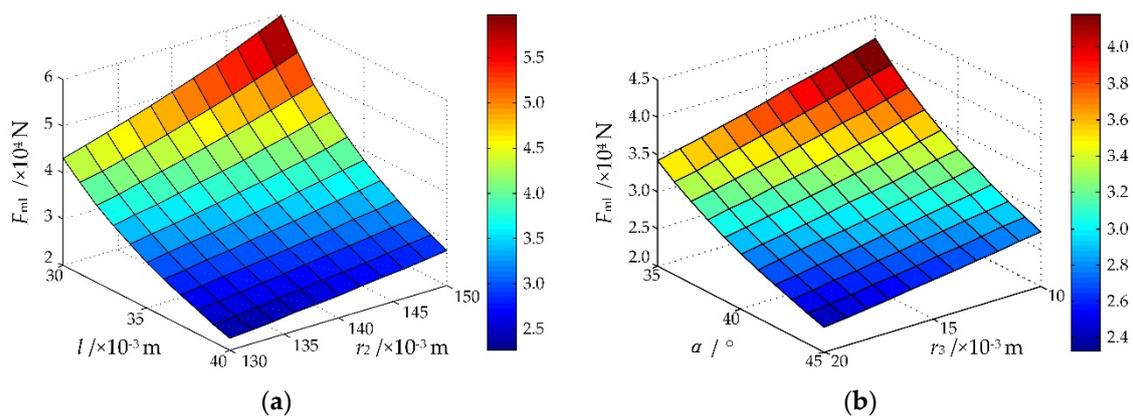


Figure 10. The lockable maximum load of the closing spring with (a) effects of r_2 and l ; (b) effects of r_3 and α .

As shown in Figure 10, F_{ml} decreases with the increase in r_3 and l . According to Equations (5), (10) and (11), both r and F_t increase with the increase in r_3 and l , resulting in the increase in the total resistance torque T . Thus, with a given output force, F_D , of the electromagnet, the maximum lockable load, F_{ml} , can be decreased by increasing r_3 and l .

Figure 11 shows the variable effects on the permissible stroke of the locking tab $[L]$. $[L]$ increases with the increase in r_1 and α or decrease in r_3 . The reason is that the angle of the notch of the locking ring, δ , increases with the increase in r_1 and α or decrease in r_3 , resulting in the rise of the rotation angle, θ , of the locking ring in the locking or unlocking process. Thus $[L]$ increases with the increase in r_1 and α or decrease in r_3 .

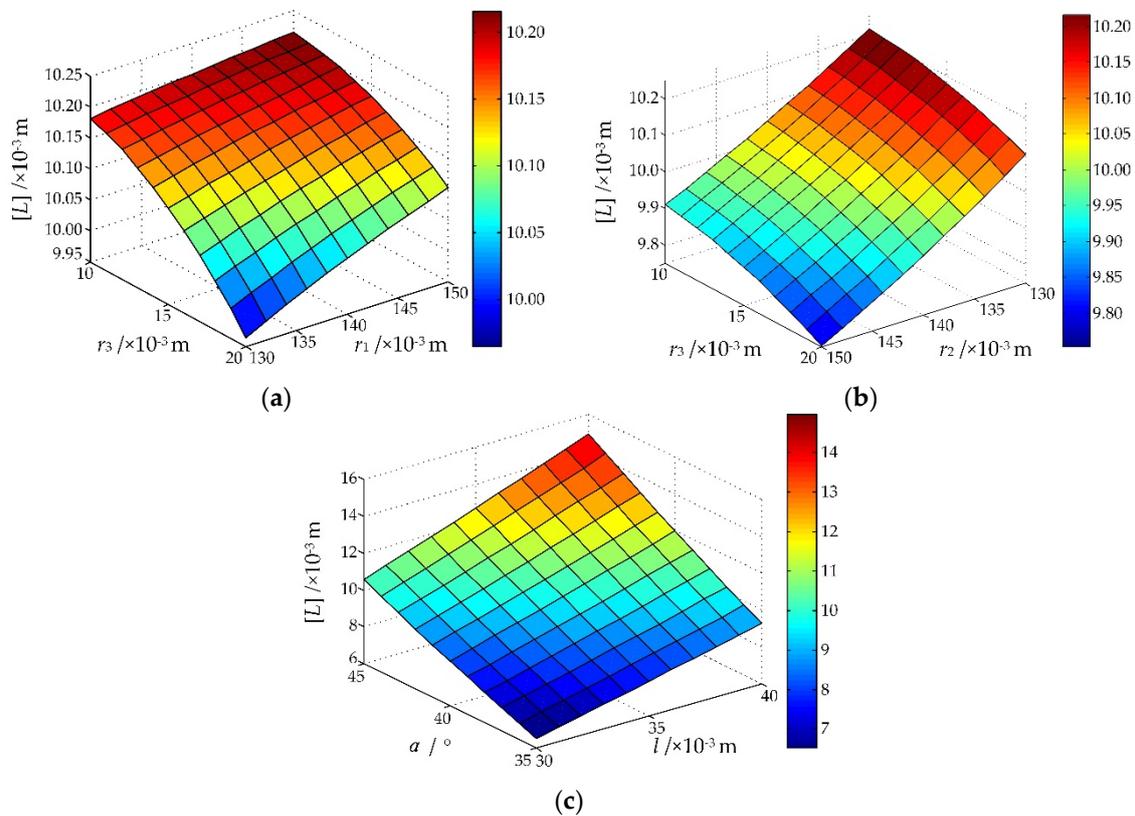


Figure 11. The permissible stroke of the locking tab with (a) the effects of r_1 and r_3 ; (b) the effects of r_2 and r_3 ; (c) the effects of l and α .

As shown in Figure 11, $[L]$ can be decreased by increasing r_2 and/or decreasing l . The reason is that as shown in Figure 9, the position of the contact point of the locking tab in the unlocking state, is not affected by the variation of r_2 and l , while the position of the contact point of the locking tab in the locking state show upward tendency with the increase in r_2 or decrease in l , resulting in the reduction in the relative position distance between the contact point in the locking state and that in the unlocking state. Thus $[L]$ decreases with the increase in r_2 or decrease in l .

4.2. Sensitivity Analysis

Sensitivity is often used to evaluate the change of performance caused by a unit variable increment. The variable effects in Section 4.1 on the objective function can be quantified by the sensitivity. The sensitivity analysis of the lockable maximum load of the closing spring F_{ml} and the permissible stroke of the locking tab $[L]$ is carried out in this section. Figures 12 and 13 show the sensitivities of F_{ml} and $[L]$ with regard to the change rate of variables in the design space, respectively, with a sampling interval of 0.01% of the full range.

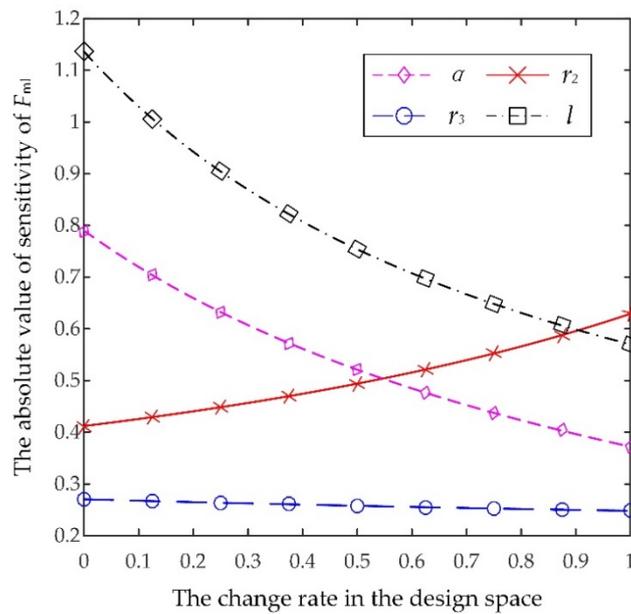


Figure 12. The absolute value of the sensitivities of F_{m1} with regard to the change rate of variables in the design space.

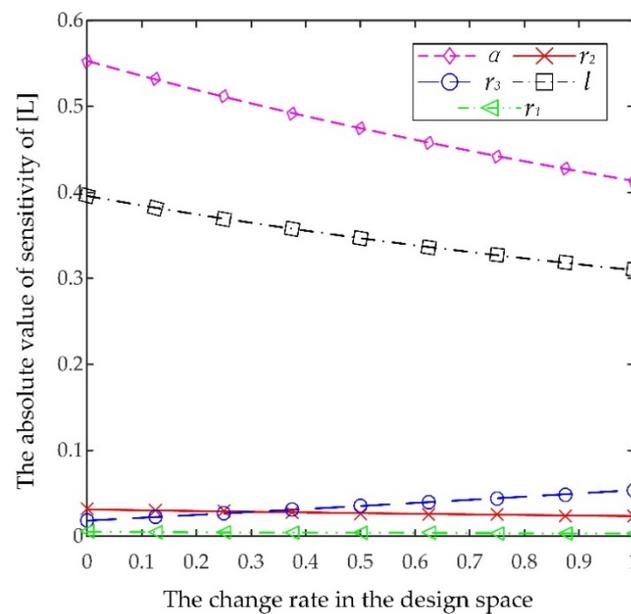


Figure 13. The absolute value of the sensitivities of $[L]$ with regard to the change rate of variables in the design space.

As shown in Figure 12, the absolute value of the sensitivities of F_{m1} increases with the increase in r_2 , and decreases with the increase in l , α and r_3 . When the change rate ranges from 0% to 54.65%, l is the most sensitive variable and after that α , r_2 , r_3 ; when the change rate ranges from 54.65% to 90.48%, l is still the most sensitive one and after that r_2 , α , r_3 ; when the change rate ranges from 90.48% to 100%, the most sensitive variable is r_2 and after that l , α , r_3 . The average variable sensitivities of F_{m1} are listed in Table 4. In the design space, the most sensitive variable of F_{m1} is l and after that α , r_2 and r_3 .

Table 4. The absolute values of the average variable sensitivities of F_{ml} in the design space.

Design Variable Description	Variable	Average Sensitivity
Radius of the notch of the locking ring	r_2	0.5027
Radius of the locking tab	r_3	0.2583
Center distance between the actuator working cavity and the locking ring notch	l	0.7852
Radial deflection angle of the center of the locking ring notch	α	0.5403

As shown in Figure 13, the absolute value of the sensitivities of $[L]$ increases with the increase in r_3 , and decreases with the increase in r_2 , l , and α ; the sensitivity of $[L]$ to r_1 remains on the lowest level in the full range; α is the most sensitive variable in the full range and after that l ; when the change rate ranges from 0% to 30.63%, r_2 is the third most sensitive variable and r_3 is the fourth. The average variable sensitivities of $[L]$ are listed in Table 5. In the design space, the most sensitive variable of $[L]$ is α and after that l , r_3 , r_2 and r_1 .

Table 5. The absolute values of the average variable sensitivities of $[L]$ in the design space.

Design Variable Description	Variable	Average Sensitivity
Radius of the actuator working cavity	r_1	0.0042
Radius of the notch of the locking ring	r_2	0.0271
Radius of the locking tab	r_3	0.0355
Center distance between the actuator working cavity and the locking ring notch	l	0.3488
Radial deflection angle of the center of the locking ring notch	α	0.4775

4.3. Analysis

According to the numerical analysis in Section 4.1, when $r_1 = 150$ mm, $r_2 = 150$ mm, $r_3 = 10$ mm, $l = 30$ mm and $\alpha = 35^\circ$, the lockable maximum load of the closing spring is $F_{ml} = 109.414$ kN and the permissible stroke of the locking tab is $[L] = 6.4059$ mm. However, the required working stroke of the locking tab $L = 8$ mm $> [L]$, which means the above variable set cannot satisfy the requirement of the low-power holding mechanism. Thus, further optimization design is needed.

According to the sensitivity analysis in Section 4.2, in the design space of the low-power holding mechanism, $[L]$ is more sensitive to l and α rather than the other variables, r_1 , r_2 and r_3 . However, F_{ml} is also sensitive to l . So α is the better option to be adjusted comparing with l to maximize the effect on the lockable maximum load of the closing spring. Through further numerical simulation, the improved parameter set of the low-power holding mechanism is obtained, as listed in Table 6, with the permissible stroke of the locking tab $[L] = 8.0031$ mm, which satisfies $[L] > L = 8$ mm, and the lockable maximum load of the closing spring is $F_{ml} = 71.44$ kN after the parameter adjustment.

Table 6. The improved parameter set of the low-power holding mechanism.

Parameter Description	Symbol	Optimal Solution
Radius of the actuator working cavity	r_1	150 mm
Radius of the notch of the locking ring	r_2	150 mm
Radius of the locking tab	r_3	10 mm
Center distance between the actuator working cavity and the locking ring notch	l	30 mm
Radial deflection angle of the center of the locking ring notch	α	39.31°

5. Conclusions

To bridge the gap of the low-power holding mechanism in the all-electric subsea gate valve actuator of the subsea production system, minimize the power consumption and cable number for

control and to improve the open-position keeping performance of all-electric subsea gate valve actuator, this paper proposed a novel low-power holding mechanism for the all-electric subsea gate valve actuator which can be applied to all-electric subsea gate valve actuators with various valve sizes and process pressure ratings. The paper also created mechanical and kinematic models, respectively, with regard to design variables, including the radius of the actuator working cavity r_1 , the radius of the notch of the locking ring r_2 , the radius of the locking tab r_3 , the center distance between the actuator working cavity and the locking ring notch l , and the radial deflection angle of the center of the locking ring notch α . Based on these analytic models, the variable effects on the lockable maximum load of the closing spring and the permissible stroke of the locking tab were analyzed. The parameter adjustment method was illustrated to satisfy the design criterion that the working stroke of the locking tab L should not exceed the permissible stroke of the locking tab $[L]$. The detailed conclusions are as follows:

- The proposed low-power holding mechanism is driven by the electromagnet and combines spiral transmission and cam-like transmission to convert the axial force of the closing spring into circumferential force to minimize the output force required as well as the number of the actuator's control cables, to improve the reliability and feasibility of the holding mechanism. The proposed low-power holding mechanism only requires a holding force of approximately 2%~7% of the maximum load of the closing spring to keep the valve open.
- With a given output force of the electromagnet, the lockable maximum load the low-power holding mechanism can be increased by increasing r_2 or decreasing r_3 , l and α . In the design space, the variable sensitivities of the lockable maximum load increases with the increase in r_2 or the decrease in r_3 , l and α ; the lockable maximum load is most sensitive to l and after that come α , r_2 and r_3 with regard to the average sensitivities in the design space.
- The permissible stroke of the locking tab increases with the increase in r_1 , l and α or the decrease in r_2 and r_3 . In the design space, the variable sensitivities of the permissible stroke increase with the increase in r_3 or the decrease in r_2 , l and α ; the sensitivity of the permissible stroke to r_1 remains on the lowest level in the full range; with regard to the average sensitivities of the permissible stroke, the most sensitive variable is α and after that l , r_3 , r_2 and r_1 .
- The design criterion is that the working stroke of the locking tab L should not exceed the permissible stroke of the locking tab $[L]$. The parameters can be adjusted to satisfy the criterion considering the variable sensitivities meanwhile minimizing the effect on the lockable maximum load. In the given case, the final parameter set of the low-power holding mechanism is: $r_1 = 150$ mm, $r_2 = 150$ mm, $r_3 = 10$ mm, $l = 30$ mm and $\alpha = 39.31^\circ$, with a lockable maximum load of the closing spring of 71.44 kN.

Further research will focus on the analysis of the influence of contact deformation and surface morphology on the modeling and the open-position keeping performance of the low-power holding mechanism. In addition, the future scope of the work can be directed toward the reliability assessment of the all-electric subsea gate valve actuator considering the subsea environment and the production fluid factors to ensure the long term application of the actuator in the subsea environment. The procedure in the paper can be applied to the design and research of other key components of the all-electric subsea production control system, such as the all-electric subsea ball valve actuator, which is required for the failsafe close ability and the open-position keeping function.

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