

Kinematic Simulation of Telescopic Sleeve Anti-Swing Device Based on Ship Swing

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Abstract. In order to reduce the impacts of wave on ship-mounted crane during operating the crane under rough weather conditions. A new type of shipboard crane telescopic sleeve anti-swing device was proposed. It uses the rigid constraints of the sleeve to reduce the swing range of the load, and through the universal torque damping mechanism to reduce stress on the telescopic sleeve. The form and principle telescopic sleeve device was introduced. A mathematical model of the ship and crane is established. Establish the simulation model by Adams, the load swing angle with having telescopic sleeve and without telescopic sleeve were achieved respectively under ship swing. At the end, the swing angle of these two models were analyzed according to the simulation results. Verify the telescopic sleeve device has a significant effect on anti-swing.

1. Introduction

When the ship is working at sea, the ship will produce rolling, pitching, yawing, swaying, surging, heaving motion in six degrees of freedom, due to the influence of wind, wave and drift. It is impossible to carry out the job when the sea conditions are badly. The anti-swing technology of the crane has a broad application prospect in the offshore hoisting operation.

On the issue of anti-swing of crane, the main research includes mechanical and intelligent anti-swing method. Maryland Rigging System^[1] established the linear dynamic model of ship crane by Lagrange method. Yuewei Li^[2] compared commonly used anti-swing mechanism. PARKER and others^[3] used the Rider Block Tagline System to control the pendulation of the load. Its structure and control are simple compared with the Maryland Rigging System. But when the rope length is longer, its function is restricted. JANG and others^[4] established the dynamic model of container ship crane, and used T-S fuzzy control method to control the swing of load. Ismail and others^[5] established the three degree of freedom model of marine rotary boom crane, and used two order sliding model to control to realize the tracking and anti-swing. Pengcheng Wang and others^[6] make dynamic analysis and established the model of marine rotary boom crane. Yongchun Fang and others^[7] designed nonlinear controller depend on the Pengcheng Wang's model and simulation tested.

Combined with the present situation and the actual situation of the vessel. This paper presents a kind of telescopic sleeve anti-swing device. Established the kinematic model of ship and crane. According to the model, analyzed the influence of the ship motion on the crane and the stress of the telescopic sleeve device. And used the Adams to do simulation experiment. Verify the telescopic sleeve device has a good effect of anti-swing.

2. Principle of anti-swing

The telescopic sleeve device is composed of a universal torque structure and a telescopic sleeve. Telescopic sleeve device is shown in Fig.1. The anti-swing principle of telescopic sleeve device is shown in Fig.2. When Crane boom hoisting and luffing by the adjustment of the hydraulic oil cylinder. The sleeve is always in a vertical downward state. When crane working, the rolling and pitching of the ship caused by waves make the load pendulation. At this time, because the rope through the telescopic sleeve device, the telescopic sleeve will bear a part of wave load, and transfer to the spherical hinge which connected with the telescopic sleeve. Three dampers of the universal torque

structure will offset the torque which transfer by spherical hinge to weaken the pendulation of the telescopic sleeve. So as to achieve the purpose of anti-swing.

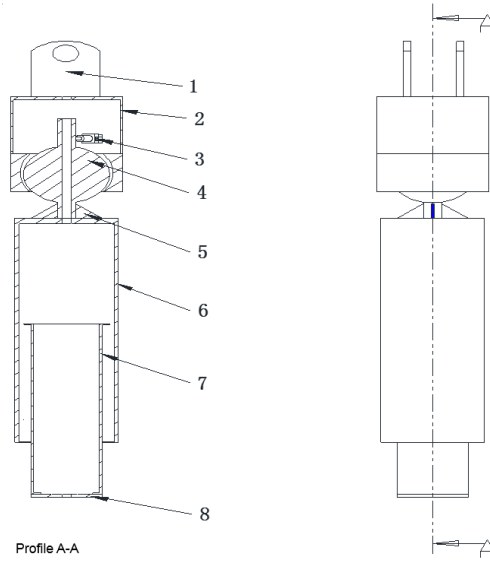


Fig.1 The structure of the telescopic casing device

1.Lug; 2.Cylinder; 3.Damper; 4.spherical hinge; 5.Connecting plate; 6.Outer pipe; 7. Inner tube; 8. End cap

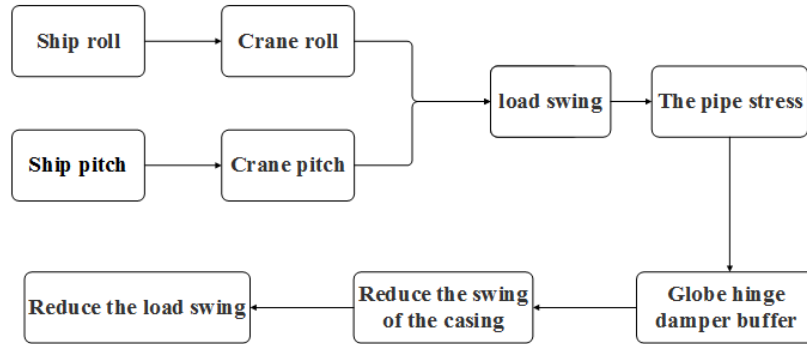


Fig.2 The anti-swing principle of the telescopic sleeve device

3. Kinematic model

In order to achieve the purpose of reducing the pendulation of the telescopic sleeve, we first establish the kinematic model of the telescopic sleeve, and use the Denavit-Hartenberg method to model the system. The models of the ship and crane, and the balance torque model of the telescopic sleeve are established.

3.1 Ship and crane model

Ship and crane model is shown in Fig.3. The corresponding connecting rod parameters are shown in Tab.1. Where $\{B\}$ is the ship coordinate, Z_B is the vertical direction, X_B is the beam direction, Y_B is the length direction, $\{0\}$ is crane base coordinate $\{1\}$, $\{2\}$ are corresponding to the crane rotary joint, variable amplitude joint, θ_1 and θ_2 respects the two joints. The origin of the coordinate $\{2\}$ corresponds to the crane lifting point, L_1 is the height of basic, L_2 is the length of crane arm.

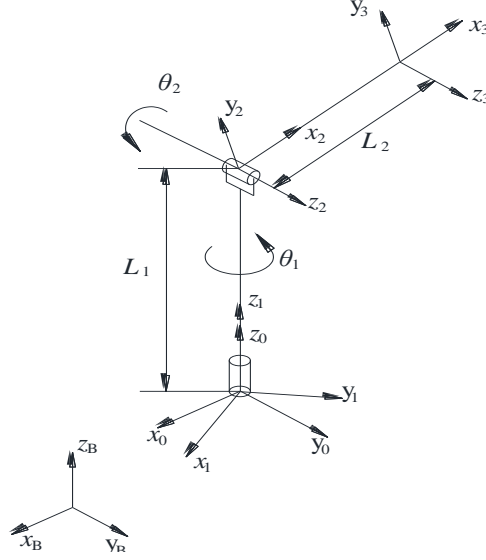


Fig.3 Ship and crane model

Table.1 connecting rod parameters

i	α_{i-1}	a_{i-1}	d_i	θ_i
1	0	0	0	θ_1
2	-90	0	L_1	θ_2
3	0	L_2	0	0

Coordinate {i} relative to the coordinate system transform for {i-1}.

$${}^{i-1}T_i = \begin{bmatrix} c\theta_i & -s\theta_i & 0 & a_{i-1} \\ s\theta_i c\alpha_{i-1} & c\theta_i c\alpha_{i-1} & -s\alpha_{i-1} & -s\alpha_{i-1}d_i \\ s\theta_i s\alpha_{i-1} & c\theta_i s\alpha_{i-1} & c\alpha_{i-1} & c\alpha_{i-1}d_i \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1)$$

The transformation matrix of the coordinate {3} relative to the coordinate {0} is:

$${}^0_3T = {}^0_1T {}^1_2T {}^2_3T \quad (2)$$

The transformation matrix of the coordinate {0} relative to the ship coordinate {B} is:

$${}^B_0T = \begin{bmatrix} 1 & 0 & 0 & B_x \\ 0 & 1 & 0 & B_y \\ 0 & 0 & 1 & B_z \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3)$$

Where (B_x, B_y, B_z) for the coordinate {0} origin in the coordinates of the ship coordinate {B}.

If the ship is located at the static sea level, the inertial coordinate {N} and the ship coordinate {B}.

When the wind and waves flow disturbed the ship rolling and pitching. The transformation matrix of the coordinate {B} relative to the inertial coordinate {N} is:

$${}^N_BT = \begin{bmatrix} \cos\alpha\cos\beta & -\sin\alpha & \cos\alpha\sin\beta & 0 \\ \cos\beta\sin\alpha & \cos\alpha & \sin\alpha\sin\beta & 0 \\ -\sin\beta & 0 & \cos\beta & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (4)$$

The α is the roll angle and the β is the pitch angle.

By Eq.2, Eq.3, Eq.4 can get the transformation matrix of coordinate {3} relate to the inertial coordinate {N}

$${}^N_3T = {}^N_BT {}^B_0T {}^0_3T = \begin{bmatrix} r_{11} & r_{12} & r_{13} & p_x \\ r_{21} & r_{22} & r_{23} & p_y \\ r_{31} & r_{32} & r_{33} & p_z \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (5)$$

Put the corresponding parameters in Table 1 to the Eq.1, Eq.2, Eq.3, Eq.4, Eq.5 can get:

$$P_x = B_x c \alpha c \beta - B_y s \alpha + B_z c \alpha s \beta - L_1 (c \theta_1 s \alpha + c \alpha c \beta s \theta_1) + L_2 (c \theta_2 (c \alpha c \beta c \theta_1 - s \alpha s \theta_1) - c \alpha s \beta s \theta_2) \quad (6)$$

$$P_y = B_y c \alpha + B_x c \beta s \alpha + B_z s \alpha s \beta + L_1 (c \alpha c \theta_1 - c \beta s \alpha s \theta_1) + L_2 (c \theta_2 (c \beta c \theta_1 s \alpha + c \alpha s \theta_1) - s \alpha s \beta s \theta_2) \quad (7)$$

$$P_z = B_z c \beta - B_x s \beta + L_1 s \beta s \theta_1 - L_2 (c \theta_1 c \theta_2 s \beta + c \beta s \theta_2) \quad (8)$$

P_x, P_y, P_z for the lifting point relative to the inertial coordinate $\{N\}$ of the roll, pitch, heave.

In order to realize anti-sway expression is expressed as:

$$\theta_1 \rightarrow -\alpha; \theta_2 \rightarrow -\beta \quad (9)$$

3.2 Torque balance model of telescopic sleeve device

Torque balance model is shown in Fig.4. X is the length direction, Y is the beam direction, Z is the vertical height. L_1 is the rod length between the upper end of the spherical joint and the damper joint. L_2 is the rod length between the upper end of the spherical joint and the end of the telescopic sleeve. F for damper resistance, G is the force of gravity, F_1, F_2, F component for G in the vertical direction and sleeve, the α is the angle between the sleeve and the Z axis, the β is the angle between the sleeve and the X axis.

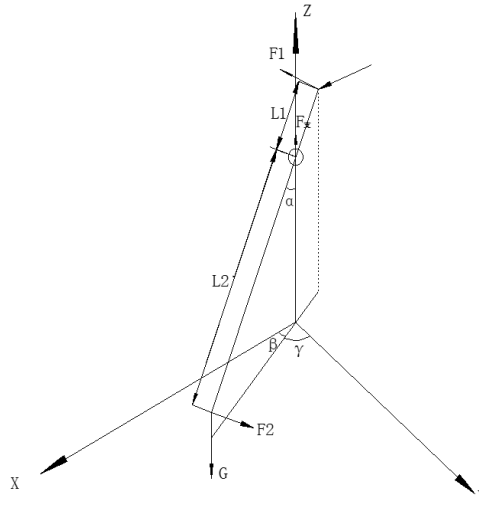


Fig.4 Torque balance model

In order to achieve the torque balance, the main vector and force of the torque are equal to zero at any point. Showed through analysis method:

$$\sum m_x(F) = 0; \sum m_y(F) = 0; \sum m_z(F) = 0 \quad (10)$$

In order to $\sum m_x(F) = 0$. So

$$F_{X1} L_{X1} = F_{X2} L_{X2} \quad (11)$$

Because the force F is provided by the damper force and it always vertical to Z axis, and the reaction force of the damper is always opposite to the sleeve movement direction. So:

$$F_{X2} = F \cos^2 \alpha \cos^2 \beta; L_{X2} = L_2 \cos^2 \alpha \cos^2 \beta \quad (12)$$

Because the gravity G is always downward, and the Tangential stress of the casing is always perpendicular to the casing. so:

$$F_{X1} = G \sin \alpha \cos \alpha \cos \beta; L_{X1} = L_1 \sin \alpha \cos \alpha \cos \beta \quad (13)$$

In summary

$$F L_2 \cos^2 \alpha \cos \beta = G L_1 \sin \alpha \cos \alpha \cos \beta; F L_1 \cos \alpha = G L_2 \sin \alpha \quad (14)$$

Similarly $\sum m_y(F) = 0$. So

$$F L_2 \cos^2 \alpha \sin \beta = G L_2 \sin \alpha; F L_1 \cos \alpha = G L_2 \sin \alpha \quad (15)$$

Similarly $\sum m_z(F) = 0$. So

$$F_Z = G \sin^2 \alpha = F_{\text{branch}} \quad (16)$$

In summary, in order to achieve anti-swing, the ship roll angle α , pitch angle β meet Eq.9, the force of telescopic sleeve device meet Eq.14, Eq.15, Eq.16.

4. Simulation analysis

Complete the three-dimensional modeling and assembly of the parts in SOLIDWORKS, then import the model into ADAMS. Create motion pairs for each joint. The simulation model is shown in Fig.5.

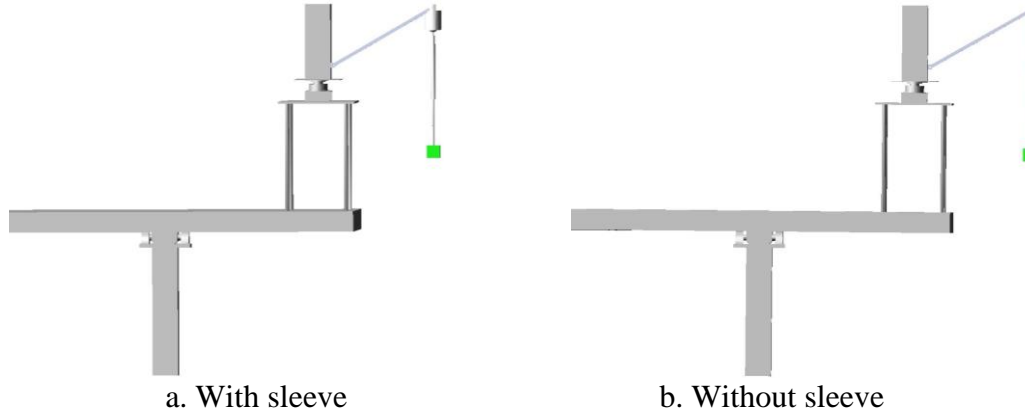


Figure.5 The simulation model

With the crane doesn't move, use the four freedom of the roll platform to simulate the movement of the ship. Set the pendulum length as 62.5 cm, 84 cm and 108 cm, and carry out simulation with sleeve and without sleeve respectively. The simulation parameters are shown in Tab. 2.

Table.2 Simulation parameters

	Group	Roll/Pitch	Pendulum length/cm	With/without sleeve
Experiment 1	E1	Roll	62.5	Without
	E2	Roll	62.5	With
	E3	Roll	84	Without
	E4	Roll	84	With
	E5	Roll	108	Without
	E6	Roll	108	With
Experiment 2	E7	Pitch	62.5	Without
	E8	Pitch	62.5	With
	E9	Pitch	84	Without
	E10	Pitch	84	With
	E11	Pitch	108	Without
	E12	Pitch	108	With

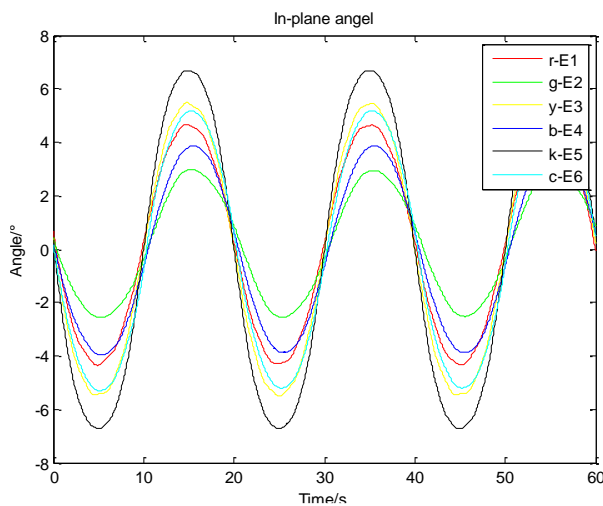


Figure. 6 In-plane angle diagram

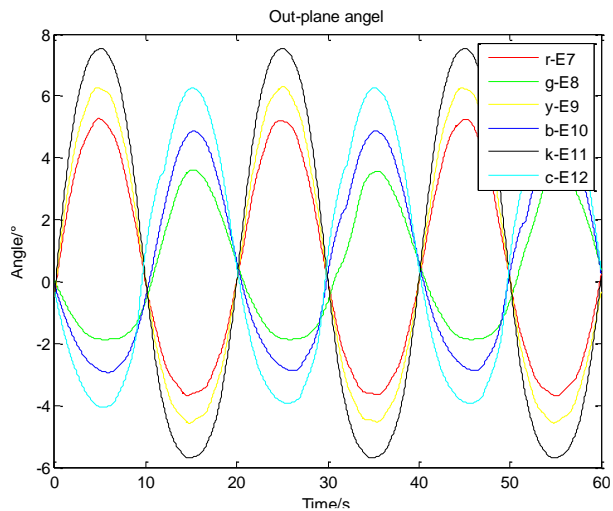


Figure.7 Out-plane angle diagram

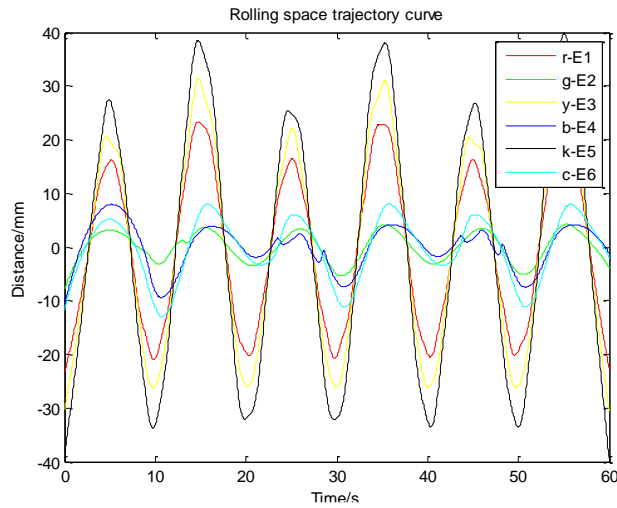


Figure.8 Rolling space trajectory curve

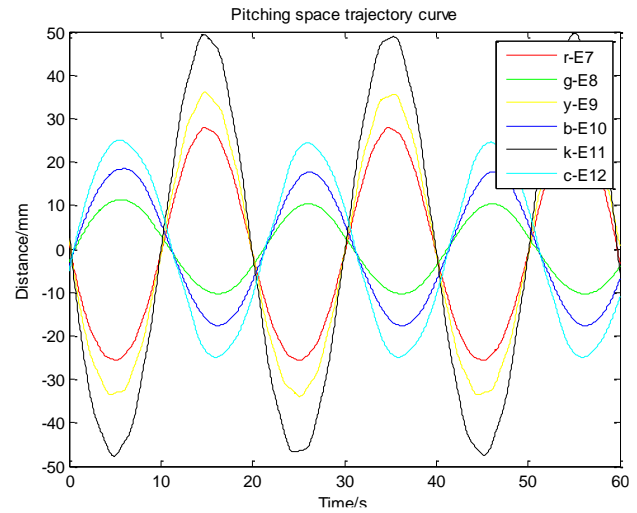


Figure.9 Pitching space trajectory curve

Get the simulation experiment results of Fig.6, Fig.7, Fig.8 and Fig.9, respectively in-plane angle and out-plane angle and space trajectory of with and without sleeve were analyzed. In roll platform rolling cases, the results shown in Tab.3 and Tab.4. In roll platform pitching cases, the results shown in Tab.5 and Tab.6.

Table.3 In-plane angle simulation result

Experiment number	Rolling/ ($^{\circ}$ s $^{-1}$)	pendulum length/cm	With/without sleeve	maximum pendulum angle/ $^{\circ}$	Reduction ratio/%
1	3.25	62.5	With	2.2	46.3
2	3.25	62.5	Without	4.1	
3	3.25	84	With	3.2	33.3
4	3.25	84	Without	4.8	
5	3.25	108	With	4.4	30.2
6	3.25	108	Without	6.3	

Table.4 Rolling space trajectory simulation result

Experiment number	Rolling/ ($^{\circ}$ s $^{-1}$)	pendulum length/cm	With/without sleeve	Maximum amplitude/mm	Reduction ratio/%
1	3.25	62.5	With	4.2	82.4
2	3.25	62.5	Without	23.8	
3	3.25	84	With	6.1	79.4
4	3.25	84	Without	29.6	
5	3.25	108	With	8.4	78.3
6	3.25	108	Without	38.7	

Table.5 Out-plane angle simulation result

Experiment number	pitching/ ($^{\circ}$ s $^{-1}$)	pendulum length/cm	With/without sleeve	maximum pendulum angle/ $^{\circ}$	Reduction ratio/%
1	3.25	62.5	With	2.5	40.5
2	3.25	62.5	Without	4.2	
3	3.25	84	With	3.8	25.5
4	3.25	84	Without	5.1	
5	3.25	108	With	4.9	21.0
6	3.25	108	Without	6.2	

Table.6 Pitching space trajectory simulation result

Experiment number	Rolling/ (° s ⁻¹)	pendulum length/cm	With/without sleeve	Maximum amplitude/mm	Reduction ratio/%
1	3.25	62.5	With	10.9	62.8
2	3.25	62.5	Without	29.3	
3	3.25	84	With	17.6	55.3
4	3.25	84	Without	39.4	
5	3.25	108	With	25.5	48.1
6	3.25	108	Without	49.1	

Compared with the rolling and pitching at each pendulation length. It can be concluded that the reduction ratio of the telescopic sleeve device is about 30%. This shows that the telescopic sleeve device has an obviously anti-swing effect. The change of the space trajectory also verify the telescopic sleeve device can meet the need work condition of shipboard crane.

5. Conclusion

Get the requirement of anti-swing and force balance relation of telescopic sleeve device through the establishment of telescopic sleeve kinematic model. Under the rolling and pitching condition of the roll platform, using Adams to simulate the telescopic sleeve device. Concluded the pendulation angle with sleeve and without angle. It can improve that the telescopic sleeve device has an obviously anti-swing effect and meet the need work condition of shipboard crane.

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