Analysis of Dynamic Characteristics of 6-PSS Parallel Mechanism Considering Spherical Hinge Clearance

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Abstract: This paper is based on a new type of 6-PSS parallel mechanism. Firstly, considering the clearance between the kinematic pair connecting the upper platform and the link, establish a kinematic model considering the spherical hinge clearance. Then, based on the Lankarani-Nikravesh(L-N) contact model and the modified Coulomb friction model calculate the contact force when the spherical elements contact, which is equivalent to the center of the corresponding member as the generalized external force. Apply for Newton Euler method with Lagrange Multiplier to establish the dynamics model of the parallel mechanism with clearance. Finally, the R-K method is used to solve the dynamic equation and analyze the influence of different spherical hinge gap sizes on the dynamic characteristics of the mechanism.

1. Introduction:
To complete the assembly and normal operation of the mechanism, it is unavoidable that there is a clearance between the motion pairs. However, the existence of the clearance will generate impact loads, affect the dynamics of the system, reduce the motion accuracy of the entire system, lead to additional vibration and affect the stability of the continuous operation of the mechanism [1]. The action mechanism of clearance can be found through the research on the clearance of the kinematics of the mechanism, and corresponding measures and strategies can be adopted to reduce the influence of the clearance of the kinematics on the mechanism and improve the performance of the mechanism.

Regarding the influence of the clearance on the dynamic characteristics of the mechanism, it is mainly concentrated on the planar mechanism and the slider mechanism [2-3]. The spatial parallel mechanism has multiple degrees of freedom and the complexity of the mechanism, and the dynamic equation dimensions and the number is large, the solution is difficult, and the research is relatively small, but some research results have also been obtained. Li [4] establishes, based on 5-PSS/UPU parallel mechanism, to consider multiple the dynamic model of the spherical hinge clearance, and the RMS error index of angular acceleration is used to quantify influence of spherical hinge clearances on structural dynamics. Wang [5] uses the improved Flores contact force model and the Coulomb friction model to establish a dynamic model considering 4-SPS/CU parallel mechanism with two clearances, and predicts the effect of the clearance mechanism on the dynamic performance.

In this paper, for a new type of space 6-PSS parallel mechanism developed, which is suitable for high speed and high acceleration motion conditions, considering the clearance between a spherical hinge connecting the upper platform and the link, a dynamic equation with Lagrange Multiplier is established based on the "contact-separation" two-state model. Using the R-K method solves the dynamic equation and analyzes the influence of different gap sizes on the dynamic characteristics of the mechanism.
2. The structure of 6-PSS parallel mechanism

2.1. The characteristics of the organization and the establishment of the coordinate system

The three-dimensional model of the 6-PSS parallel mechanism is shown in Fig 1. The mechanism mainly consists of an upper platform, a lower platform, and six branch chains. Each branch chain is composed of motor, reducer, lead screw nut, guide rail, slider and link. Because the mechanism arranges the entire driving mechanism horizontally on the lower platform, it improves the carrying capacity of the entire mechanism and the stability and enlarges the working space of the upper platform. At the same time, the six guide rails are arranged in an inner and outer equilateral triangle, which can make the whole mechanism more compact and reduce the layout space.

![Fig 1. Model of 6-DOF mechanism](image1)

Fig 1. Model of 6-DOF mechanism  
Fig 2. Schematic diagram of 6-DOF mechanism

Fig 2 shows a schematic diagram of the parallel mechanism when it is in the neutral position (initial position). Establish coordinate systems on the upper and lower platforms respectively. \( D \) and \( O \) are the centers of the upper and lower platforms, and \( O – XYZ \) is the coordinate system fixed to the center of the lower platform. The \( Z \) axis is perpendicular to the plane of the lower platform and the \( X \) axis is connected to one of the horizontal guide rails. The \( Y \) axis is determined by the right-hand spiral theorem. \( D – XYZ \) is a coordinate system fixedly connected to the center of the upper platform, and the direction of the coordinate system is the same as the direction of the fixed coordinate system when the entire mechanism is in the neutral position. Six guide rails are arranged on the lower platform to form two inner and outer equilateral triangles. The circumscribed circle radius is \( R_1 \) and \( R_2 \) respectively. The six spherical hinge points on the upper platform are distributed on a circle with a semi-diameter of \( r \). The lengths of the inner and outer links connecting the upper and lower spherical hinges are \( l_1 \) and \( l_2 \) respectively.

2.2. Inverse solution of mechanism kinematics

According to the coordinate rotation formula, the rotation matrix of the upper platform coordinate system relative to the fixed platform coordinate system is:

\[
\rho_{\alpha \beta \gamma} = D_{OD} \rho_{\alpha \beta \gamma} R_{Y} R_{Z} R_{X}
\]

Then the coordinates of the spherical hinge point on the upper platform under fixed coordinates can be expressed as:

\[
A_i = D + \rho_{\alpha \beta \gamma} R_{X} R_{Y} R_{Z} A_i
\]

\( \rho_{\alpha \beta \gamma} \) is the coordinate of the upper spherical hinge point \( A \) in the coordinate system of upper platform. The angle between the directions of the six guide rails and the \( X \) axis of the fixed coordinate system is \( \varphi \), and set the coordinates of the guide rail vertex \( C_i = (x_i, y_i, 0) \), then the coordinates of the lower spherical hinge: \( B_i = (x_i + \rho_{\alpha \beta \gamma} \cos \varphi, y_i + \rho_{\alpha \beta \gamma} \sin \varphi, 0) \).

The distance between the upper and lower spherical hinges is equal to the length of the link, which can gain the inverse solution of the mechanism.

3. Kinematics model considering spherical hinge clearance

For an ideal sphere hinge, the geometric centers of the motion pair connecting the sphere and the sphere socket are completely coincident, and the sphere socket rotates in three directions in the middle of the sphere. In fact, there is a clearance between the elements that make up the sphere hinge, which causes the sphere to move in three directions in the sphere socket. The motion constraint is transformed into a
force constraint. Due to the existence of the clearance, the centers of the two do not coincide.

![Fig 3. Clearance spherical hinge model diagram](image1)

Fig 3. Clearance spherical hinge model diagram

Fig 4. Branch chain closed loop vector diagram

The kinematic model of the sphere hinge considering the clearance is shown in Fig 3. The sphere centers of the sphere socket and the sphere are \( O_1 \) and \( O_2 \). The eccentric vector is \( e \), of which direction is from the center of the sphere socket to the center of the sphere. When the sphere socket and the sphere are in contact with each other, suppose the contact points are \( P_1 \) and \( P_2 \) respectively, and contact deformation is \( \delta \). The normal and tangential unit vectors of the contact surface are \( \mathbf{n} \) and \( \mathbf{t} \) respectively.

![Fig 4. Branch chain closed loop vector diagram](image2)

Fig 4 is a vector diagram of a branched-chain closed loop considering a sphere hinge with clearance. \( B \) is the center position of the lower sphere hinge, and \( C \) is the position of the top of the guide rail. The eccentric vector between the sphere socket and the sphere can be obtained through Fig 3 and 4:

\[
e = O_2O_1 - O_2O_1 = O_2D + \theta_2R_2 \mathbf{A}_2 \quad O_2O_1 = OC + CB + BO_2
\]

(2)

Then the eccentric unit vector between the sphere and the sphere socket is \( \mathbf{e}_n = e/|e| \), \( |e| \) is the modulus of vector \( e \). Assuming that the radii of the sphere socket and the sphere are \( r_1 \) and \( r_2 \), the sphere hinge clearance is \( \text{gap} = r_1 - r_2 \), and the contact deformation is \( \delta = |e| - \text{gap} \). This paper uses the "contact-separation" state model, assuming that there are only two states of contact and separation between the sphere socket and the sphere. Therefore, it can be judged whether the sphere socket and the sphere are in contact according to the contact deformation at two adjacent moments:

\[
\begin{align*}
\text{when } & \delta_{i+1} - \delta_i > 0, \text{ Continuous contact and degree of freedom state} \\
\text{when } & \delta_{i+1} - \delta_i = 0, \text{ Just touched or separated}
\end{align*}
\]

(3)

When contact occurs, suppose the contact points of the sphere socket and the sphere are \( P_1 \) and \( P_2 \), respectively, then:

\[
OP_1 = O_1D + \theta_1R_1 \mathbf{A}_1 + r_1n_1 \quad OP_2 = OC + CB + BO_2 + r_1n_1
\]

(4)

Take the derivative of (6) to get the contact velocity between the sphere socket and the sphere:

\[
\mathbf{\dot{OP}}_1 = \dot{O}_1D + \theta_1\dot{R}_1 \mathbf{A}_1 + \dot{r}_1 \mathbf{n}_1 \quad \mathbf{\dot{OP}}_2 = \dot{O}_2D + \theta_2\dot{R}_2 \mathbf{A}_2 + \dot{r}_2 \mathbf{n}_2
\]

(5)

The normal and tangential velocities can be obtained by projecting formula (5) to the contact surface and the normal plane of the contact surface respectively:

\[
\mathbf{v}_n = [(\mathbf{\dot{OP}}_1 - \mathbf{\dot{OP}}_2) \mathbf{n}] \mathbf{v}_n \quad \mathbf{v}_t = (\mathbf{\dot{OP}}_1 - \mathbf{\dot{OP}}_2) - \mathbf{v}_n
\]

(6)

Then the tangential unit vector of the contact surface is:

\[
\mathbf{n}_t = \mathbf{v}_t/|\mathbf{v}_t|
\]

4. Consider the contact force of the sphere hinge clearance

4.1. Normal contact force

The classic Hertz contact model regards the contact problem as a completely elastic contact, ignoring the damping, and does not consider the energy loss during the contact process. The Lankarani-Nikravesh contact model (L-N model) proposes a nonlinear damping model that considers the coefficient of restitution, introduces the initial contact velocity and material properties \(^{[7]}\) and takes into account the energy change during the contact process, which is closer to the real situation. Therefore, this paper uses the L-N contact model to describe the contact force of the spherical secondary elements of the parallel mechanism:
\[ F_n = K \delta^n [1 + 3(1 - c_r^2) \dot{\delta}^3/(4 \dot{\delta}_0)] \]  

Where \( n \) is the index; \( c_r \) is the recovery coefficient of the spherical hinge; \( \dot{\delta} \) and \( \dot{\delta}_0 \) are the contact deformation velocity and the initial contact deformation velocity, respectively; \( K \) is the stiffness coefficient:

\[ K = 4/(3\pi(\sigma_c + \sigma_s)) \sqrt{r_1 r_2/(r_1 + r_2)} \cdot \sigma_c = (1 - \mu_c^2)/\pi E_s; \quad \sigma_s = (1 - \mu_s^2)/\pi E_s \]  

Where \( E_s \) and \( E_s \) is Young’s modulus of the sphere socket and sphere; \( \mu_c \) and \( \mu_s \) is Poisson’s ratio of the sphere socket and sphere respectively.

4.2. Tangential contact force
Since the surface of the sphere socket and the sphere is not smooth, friction is unavoidable. The most classic friction model at present is the Coulomb friction model, which expresses the friction as the product of the positive pressure and the friction factor. In order to prevent the frictional force from becoming discontinuous due to the direction change when the tangential velocity is near zero, the Coulomb friction model with correction coefficient is used to describe the frictional force of the spherical hinge in the contact process:

\[ F_c = -s \mu c F_n \]  

Where \( s \) is the coefficient of friction, which is 0.1; \( c \) is the amended coefficient of friction.

\[ c = \begin{cases} 0 & \text{if } |v_t| < v_0 \\ \left|v_t/v_0 - v_0\right|/(v_1 - v_0) & v_0 \leq |v_t| \leq v_1 \\ 1 & |v_t| > v_1 \end{cases} \]  

Where \( v_0 \) and \( v_1 \) are specific limit velocity within the error range. \( v_0 = 0.01 \text{m/s}, v_1 = 0.1 \text{m/s} \).

4.3. Equivalence of contact force
According to the normal and tangential contact force models established above, the contact force is obtained:

\[ F_n = F_{n,S} + F_{n,L} \]  

Force and torque can be used to equate the contact force to the center of the upper platform and the link:

\[ F_{n,S} = -F_3; \quad M_{n,S} = (OP - OQ)F_3; \quad F_{n,L} = F_3; \quad M_{n,L} = (OD - OP)F_3 \]  

\( F_{n,S} \) and \( M_{n,S} \) are the force and torque equivalent to the center of the link, respectively. \( F_{n,L} \) and \( M_{n,L} \) are the force and torque equivalent to the center of the upper platform, respectively.

5. Dynamic model considering the clearance
5.1. Mechanism kinematics constraint equation
Each component of the 6-PSS parallel mechanism can be represented by generalized coordinate \( q_i = (x_i, y_i, z_i, \gamma_i, \beta_i, \alpha_i) \), where \( x_i, y_i, z_i \) are the coordinates of the component center in the fixed coordinate system, and \( \gamma_i, \beta_i, \alpha_i \) are the rotation angles of the component coordinate system around the fixed coordinate. And the generalized coordinate of the moving pair composed of the guide rail and the sliding block can be expressed by the displacement, the generalized coordinate of the whole mechanism: \( q = (s_i, q_{16}, q_{17}) (i = 1 - 6) \)

When the link rotates around its own axis, it does not affect the movement of the entire mechanism. Therefore, in order to simplify the kinematics constraint equation. Assuming that the angle of the link turning around its own axis is zero, the generalized coordinate of the link becomes \( q_i = (x_i, y_i, z_i, 0, \beta_i, \alpha_i) \). Assume the direction vector of the link \( L_i = (x_i, y_i, z_i) \). \( \beta_i \) and \( \alpha_i \) can be obtained by the direction vector of the link.

\[ \beta_i = \arccos(z_i/|L_i|); \quad \alpha_i = \arctan(y_i/x_i) \]  

The topology diagram of the mechanism can easily and intuitively see the relationship between each
kinematic pair and the components connected to the kinematic pair, which helps to quickly and accurately establish the kinematic constraint equation of the mechanism. The topology diagram of the 6-PSS parallel mechanism is Fig 5, the numbers 0-13 respectively represent the lower platform, 6 sliders, and 6 links, and the numbers H1-H18 represent various kinematic pairs that connect these components. The points in parentheses indicate the center of the coordinate system.

Fig 5. Topology diagram of 6-pss parallel mechanism Fig 6. Model diagram of spherical hinge

The component is represented by the center point of the component coordinate system. A kinematic pair corresponds to the center point of the two components. The center points of the two components can be regarded as a group of associative arrays of the kinematic pairs, and the associative array is in turn related to the topology of the 6-PSS parallel mechanism. There is a one-to-one correspondence between the graphs. The associative array of the 6-PSS parallel mechanism can be shown in Tab1. Since this paper does not need to establish a coordinate system at the center of the slider, the center of the slider is represented by "."

Tab1. Associative array of 6-PSS parallel mechanism

| i | j |
|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
| O | O | O | O | O | O | O | O | O | O | O | O | O | O | O | O | O | O |

| i | j |
|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 |
| - | - | - | - | - | - | - | - | O | O | O | O | O | O | O | D | D | D | D |

The generalized coordinates of the moving pair are represented by only one degree of freedom, so there is no need to establish the constraint equation of the moving hinge, only the constraint equation at the spherical hinge is considered. The model of the spherical hinge is shown in Fig 6.

![Fig 5. Topology diagram of 6-pss parallel mechanism](image1)

The constraint equation of spherical hinge:

\[ \Phi(q_i, q_j) = r_i + \ell_i^o R_i P - (r_j + \ell_j^o R_j P) = 0_{18} \] (12)

The constraint equation of the spherical hinge connecting the slider and the link of the 6-PSS parallel mechanism:

\[ \Phi(q_i, q_o) = r_i + \ell_i^o R_i P - (r_o + \ell_o^o R_o P) = 0_{18} (i = 1-6) \]

The constraint equation of the spherical hinge connecting the upper platform and the link of the 6-PSS parallel mechanism is:

\[ \Phi(q_i, q_o) = r_i + \ell_i^o R_i A - (r_o + \ell_o^o R_o A) = 0_{18} (i = 6-12) \]

In order to make the mechanism have a definite movement, a set of driving constraints must be imposed on the mechanism. The driving constraint equation is the slider displacement obtained by the inverse solution of the platform pose on the known mechanism, which can be expressed as:

\[ \Phi(q_s, q_i) = s_i - [B_i C_i] = 0_{18} (i = 1-6) \]

Synthesizing the constraint equations of the spherical hinges and, the constraint equations of the 6-PSS parallel mechanism are:
\[ \Phi(q,t) = (\Phi(q_i, q_{i+1}) \hat{q}, \Phi(q_{i+1}, q_{i+2}) \hat{q}) \cdot \Phi(q_{i+2}, q_{i+3}) \hat{q} = 0, \quad (i = 1, 2) \]  

(13)

Since there is a clearance at the sixth spherical hinge connecting the upper platform and the link considered in this paper, the motion constraint is transformed into a force constraint, which needs to be removed from the ideal kinematic constraint equation. The removed kinematic constraint equation: \( \Phi(q_6, q_7) \hat{q} = 0 \)

Derivation of the kinematics constraint equation (13) and sorting it out can get the velocity and acceleration constraint equation:

\[ \dot{\Phi}/\ddot{q} \cdot \dot{q} + \dot{\Phi}/\dddot{q} = 0; \quad \gamma = \Phi \dot{q} = -\left( \dddot{\Phi}/\dddot{q} \cdot \dddot{q} + 2 \cdot \dddot{\Phi}/\dddot{q} \dddot{q} + \dddot{\Phi}/\dddot{q} \right) \]  

(14)

5.2. Establish the clearance dynamic equation

According to Newton's Euler formula with Lagrange multipliers, a dynamic equation including the kinematic constraints of the mechanism can be established without considering the clearance:

\[ \begin{bmatrix} M & \Phi_x^T & q \\ \Phi_x & 0 & \lambda \end{bmatrix} = \begin{bmatrix} F \\ \gamma \end{bmatrix} \]  

(15)

where \( \lambda \) is Lagrange multiplier; \( M \) and \( F \) are the generalized mass matrix and generalized external force, respectively.

When considering the sphere hinge clearance, remove the kinematic constraint equation of the spherical hinge connecting the upper platform and the link, and the contact force at the spherical hinge is equivalent to the center of the corresponding member, then the generalized force received of the upper platform and the sixth link:

\[ F_x = F_{x1} \quad F_{x2} \quad F_{x3} \quad M_{x1} \quad M_{x2} \quad M_{x3} \]  

\[ F_y = F_{y1} \quad F_{y2} \quad F_{y3} \quad M_{y1} \quad M_{y2} \quad M_{y3} \]  

(16)

The position and velocity equations can only meet the requirements at a certain discrete moment, but the acceleration equation has a default phenomenon, which makes it impossible to obtain stable numerical solutions through numerical method. Baumgart algorithm [9] is used the most commonly solution. In the Baumgart algorithm, the velocity and position constraints are introduced into the acceleration term. Then the kinetic equation (15) becomes:

\[ \begin{bmatrix} M & \Phi_x^T & q \\ \Phi_x & 0 & \lambda \end{bmatrix} = \begin{bmatrix} F \\ \gamma - 2a \Phi - b \Phi \end{bmatrix} \]  

(17)

Where \( a \) and \( b \) are default correction coefficients. When \( a \) and \( b \) are positive values, the system can generally reach stability, and when \( a = b \), the system can quickly reach a stable state.

6. Numerical simulation

This paper uses R-K method to analyze the dynamic characteristics of the mechanism. The entire solution flow chart is shown in Fig 7.

Fig 7. Flow chart of clearance calculation

In order to quantitatively evaluate the influence of different gap sizes on the dynamic characteristics of the mechanism, this paper selects the mean square root error of acceleration of the upper platform as the quantitative evaluation index of the simulation results.
Where \( \sigma \) is the mean square root error of acceleration; \( n \) is the sample size; \( \ddot{q}_i \) and \( \ddot{q}_i \) represent the acceleration with the clearance and the ideal acceleration, respectively. This chapter mainly analyzes the effect of the gap size on the dynamic performance of the mechanism through numerical simulation. The clearance parameters used are shown in Tab 2.

### Tab 2. Clearance parameters

| radius (m) | Poisson's ratio | Elastic Modulus (Pa) | Coefficient of restitution |
|------------|-----------------|----------------------|---------------------------|
| sphere socke | 0.0402           | 0.29                 | 2.07x10^11                | 0.9                        |
| sphere     | 0.0400           | 0.29                 | 2.07x10^11                | 0.9                        |

The paper mainly analyzes the impact on the dynamic performance of the mechanism when there is a clearance at the upper spherical hinge H18, and the gap size \( \text{gap}=0.01\text{mm}, \ 0.05\text{mm}, \ 0.1\text{mm} \) and \( 0.2\text{mm} \). The simulation parameters of the mechanism are shown in Tab 3. Because the mechanism is under the action of the single kinematic pair clearance, the upper platform's Y-direction and Z-direction movement laws are similar, so this paper uses the upper platform Z-direction movement parameters to illustrate the influence of the kinematic pair clearance on the dynamic characteristics of the entire mechanism. Given the center movement curve of the upper platform and kinetic simulation parameters Tab 3:

\[
Y = 0.12\sin(4\pi t) \quad Z = 0.7420 + 0.12\cos(4\pi t)
\]  

### Tab 3. Kinetic simulation parameters

| Inscribed circle radius of inner rail(m) | \( R_1 = 0.5915 \) | Inscribed circle radius of outer guide rail(m) | \( R_2 = 0.8815 \) | Median height(m) | \( H = 1.2200 \) |
|-----------------------------------------|--------------------|-----------------------------------------------|--------------------|------------------|------------------|
| Upper spherical hinge radius(m)         | \( r = 0.7000 \)   | Inner link length(m)                          | \( l_1 = 1.0411 \) | Length of outer link(m) | \( l_2 = 1.1964 \) |
| Slider quality(kg)                      | \( m_s = 3.4 \)     | Short link quality(kg)                        | \( m_{s1} = 5.0 \) | Platform quality(kg) | \( m_s = 20.0 \)   |
| Long link quality(kg)                   | \( m_{l2} = 5.9 \)  |                                               |                     |                  |                  |
| Short link inertia(kg.m^2)              | \( j_s = j_s = 0.1478 \) | \( j_s = 0.000498 \)                        | \( j_s = j_s = 0.17745 \) | \( j_s = 0.0005771 \) |
| Long link inertia(kg.m^2)               | \( j_s = j_s = 0.5271 \) | \( j_s = 1.0542 \)                           |                     |                  |                  |
| Upper platform inertia(kg.m^2)          | \( j_s = j_s = 0.5271 \) | \( j_s = 1.0542 \)                           |                     |                  |                  |

From Fig 8(a) of displacement with different gap sizes, it can be seen that the Z-direction displacement curve with clearance is relatively smooth and basically coincides with the ideal displacement curve, indicating that the clearance has little effect on the displacement accuracy of the upper platform. Further from the displacement error Fig 8(b), it can be seen that the absolute value of the maximum displacement error in the Z direction increases with the increase of the gap size. When the gap size changes from 0.01mm to 0.2mm, the Z-direction displacement error is that about 0.005 mm rises to 0.09mm, and the maximum displacement error is less than the gap size.

From Fig 9, it can be seen that the velocity with clearances becomes unsmooth, with many burrs appearing, and fluctuates up and down the ideal curve, and it is most obvious near the velocity extreme. It can be seen from the partial enlarged view of the velocity that the amplitude of the fluctuation increases with the increase of the gap size.

From Fig 10, it can be seen that there are many spikes in the acceleration curve under different gap sizes. The value of the maximum spike increases with the increase of the gap size. When the clearance ranges from 0.01mm to 0.2mm, The Z direction ranges from 27.6 m/s^2 to 60.2 m/s^2, which is due to the increase in the maximum contact force as the gap size increases and leads to a corresponding increase.
in the maximum acceleration.

Fig 10. Z-direction acceleration with different gap sizes

Finally, increase some gap sizes and use the mean square root error of acceleration proposed to quantify the degree of influence of different gaps on the dynamic characteristics of the upper platform. It can be seen from Tab 4 that mean square root error of acceleration increases with the increase of the gap size, which decreases dynamic performance and stability; when the gap is 0.01mm and 0.02mm, mean square root error of acceleration slightly changes, which means that it is of little significance to improve the dynamic performance of the mechanism in reducing the gap size which increases the cost when the gap size is 0.02mm.

| clearance(mm) | 0.01 | 0.02 | 0.05 | 0.1  | 0.2  | 0.3  | 0.5  |
|--------------|------|------|------|------|------|------|------|
| Z(m/s^2)     | 2.5693 | 2.5877 | 4.0158 | 5.4995 | 7.5047 | 11.4286 | 14.3850 |

7. Conclusion

Based on a new type of 6-PSS space parallel mechanism, this paper uses Newton’s Euler formula with Lagrange Multiplier to establish a dynamic model with spherical hinge clearance, and analyzes the influence of different gap sizes on the dynamic characteristics of the entire mechanism. Draw the following conclusion, different gap sizes have a little effect on displacement and velocity, but have greater influence on acceleration and contact force. The increase of the gap size will deteriorate the dynamic characteristics of the mechanism and decrease the stability.

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