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Modeling and Analysis of the Flap Actuation System Considering the Nonlinear Factors of EMA, Joint Clearance and Flexibility

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Abstract: The performance of the flap actuation system directly affects the control effect and the flight quality of an aircraft. The electromechanical actuator (EMA) and the linkage mechanism are important components of the system. In order to achieve the goals of good transmission accuracy and dynamic response, the influence of nonlinear properties in the transmission chain including the EMA and linkage mechanism should be considered. A co-simulation model at the system-level of the flap actuation system was developed, which takes nonlinear factors of the EMA, the impact dynamics of the linkage mechanism with joint clearance and the rigid–flexible coupling characteristics into account. Moreover, the experiments with different command frequencies and loads were performed. The simulation and experimental results were compared to verify the effectiveness of the co-simulation model. Finally, the effects of nonlinear properties including the contact stiffness and clearance of a planetary roller screw mechanism, EMA anchorage stiffness, number of clearance joints, flexibility and load are discussed. This work can contribute to analyzing the performance of an electromechanical multibody system with nonlinear characteristics, which has crucial academic meaning and engineering application values for the development of systems with high speed, good reliability and long life.



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Keywords: flap actuation system; electromechanical actuator; co-simulation model; joint clearance; flexibility

1. Introduction

The fly-by-wire flight control system has been extensively employed to ameliorate the maintainability, controllability and reliability of aircrafts. One of the key components of it is the flap actuation system, which is composed of two subsystems, namely, an electromechanical actuator (EMA) and a linkage mechanism.

The actuator is an important part of the flap actuation system, which can be divided into pneumatic, hydraulic and electric types according to the driving mode. The EMA has advantages such as light weight, high efficiency, easy maintenance and easy installation, but it also has disadvantages such as its performance being easily affected by energy, loads and nonlinear factors [1–3]. The EMA is a multibody system consisting of the motor, controllers, reducer, actuator and sensors, in which the clearance, friction and other nonlinear factors are detrimental to the output characteristics of the system. The research on EMAs mainly includes motor control [4], fault diagnosis [5–8] and dynamic characteristic analysis. In terms of the dynamic modeling and analysis of the EMA, Maré and Fu [9–12] conducted the modeling of clearance, friction and flexibility for the EMA and elaborated on the top-down

parameterized design method of EMA position control. Merzouki et al. [13] established a disturbance model of nonlinear factors for the mechanical transmission part of the EMA and improved the control performance through an adaptive algorithm. One of the focuses in this paper is to develop the EMA model with multiple nonlinear factors and analyze their effects on the output responses in this paper.

The linkage mechanism can realize the movement and power transmission between the EMA and control surface. The collision effects of the clearance joints of the mechanism will reduce the transmission precision and enhance the noise and vibration. In engineering, the clearance is inevitable. On the one hand, a reasonable fit clearance can meet the relative motion between adjacent components. On the other hand, errors of manufacturing and assembling are inevitable [14]. And the wear between the components of the motion pair will further increase the clearance size [15]. Li et al. [16] proposed a feasible simulation model of the air rudder mechanism with joint clearance. Bai et al. [17] investigated the effects of joint clearances on the dynamic responses of the dual-axis driving mechanism. Zhuang et al. [18] took a dual-axis driving mechanism as an object and proposed an efficient wear prediction method. Li et al. [19] built a multi-clearance dynamics model of the deployable solar arrays and found that the collision aggravated the satellite yaw and made the vibration lag. Guo et al. [20] proposed a contact force model suitable for small clearances in revolute joints and conducted the research and experimental verifications on the dynamic characteristics of the rudder loop system. At present, many studies about aircrafts and spacecraft focus on the modeling of the transmission mechanism with joint clearance and flexibility. In addition, research on the dynamics of mechanisms with clearances includes the analysis of the wear [21], lubrication [22,23], coating [24] and uncertainty parameters [25,26]. However, fewer investigations of the coupling effects of the actuator and linkage mechanism have been carried out.

The multibody system consists of many components and is connected by several joints. Flores et al. [27] presented a rigid simulation model with clearance joints and indicated that the responses were influenced by the clearance size and the number of clearance joints. Tian et al. [28] established a new model for analyzing a planar flexible system with clearance joints based on the absolute node coordinate method. The results showed that the contact force of the flexible model was far less than that of a rigid one. Ma et al. [29] analyzed dynamic behaviors of a slider-crank mechanism with clearances. The results showed that the motion mode in one clearance joint produced a significant influence on that of the other joint. Tan et al. [30] studied the responses of a slider-crank mechanism with multi-clearance joints and indicated that the responses with two clearance joints were not simply superimposed on those with one. As mentioned above, the influence of multiple clearance joints on the system behaviors cannot be overlooked.

After conducting extensive research, the effects of flexibility are investigated. Dubowsky et al. [31] proved that the rod's flexibility could diminish the contact force. Schwab et al. [32] indicated that the key to solve the peak contact force was the dynamics model used and that the flexibility could make the contact force curve relatively smooth. Xiao et al. [33] considered coupling effects of the clearance joint and link's flexibility and developed a rigid-flexible model of the multi-link mechanism. The responses analysis and chaos identification were performed through phase diagrams, Poincare diagrams and the maximum Lyapunov exponent. Song et al. [34] found that when the link's flexibility was considered, the movement accuracy of a flexible parallel robot was reduced and the motion stability of the mechanism with flexible links was better than that only considering clearance joints. Jiang et al. [35] proposed a modeling method for a multi-link mechanism considering the coupling effects of the clearance and flexibility, pointing out that the flexibility can effectively reduce the peaks of dynamic responses. Shen et al. [36] established a dynamic model of a solar array system with flexible panels and clearance joints. Through unfolding and locking process simulations, it was shown that the equivalent suspension damping characteristic generated by the coupling effects of the flexibility and clearance could reduce the vibration of the deployable component. López-Lombardero et al. [37]

described the flexibility of the connecting rod for a slider-crank mechanism using the floating coordinate method and pointed out that the flexibility could reduce the adverse effects of the clearance and wear. In view of these, studying the effects of the component's flexibility on the system's behaviors is certainly worthwhile.

The structure of a flap actuation system is represented in Figure 1. Actuator control electronics (ACE) transforms the command into the power to drive a permanent magnet synchronous motor (PMSM). Then, the PMSM speed is decelerated and the torque is increased through a reducer. Next, the rotary motion is converted into linear motion by a planetary roller screw mechanism (PRSM) and the linear displacement and force are output to drive the lever arm rotation. Finally, the deflection of the control surface of an aircraft is accomplished. The responses of the flap actuation system are not only related to the reproduction ability of the EMA to the command signal but are also relevant to the dynamic characteristics of the linkage mechanism. Therefore, it is important to consider the coupling effects of nonlinear factors of the EMA, the collision dynamics of clearance joints and the rigid-flexible coupling characteristics of the linkage mechanism. Establishing a co-simulation model is an effective way to comprehensively analyze the characteristics of the flap actuation system. This is significant for the structural design and performance analysis of the aircraft.

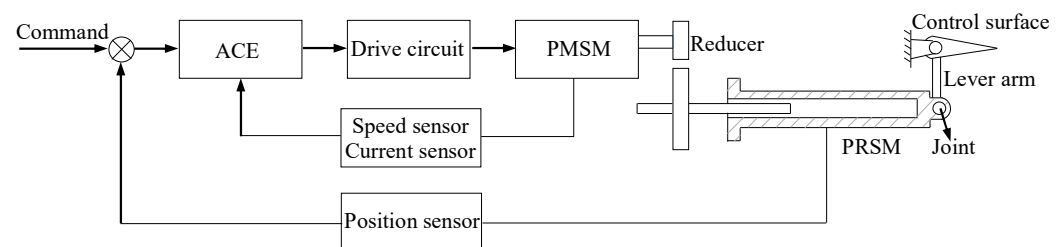


Figure 1. Structure composition of the flap actuation system.

First, mathematical models including contact force models and dynamics equations of the flap actuation system are given. Next, the rigid-flexible model of the linkage mechanism and the dynamics model of the EMA with the three-closed-loop control are established, respectively. Then, based on the interaction of the variables between those two models, a system-level co-simulation model is developed. Finally, the comparison between the simulations and experiments is conducted to verify the effectiveness of the co-simulation model and the influence of the nonlinear properties existing in the transmission chain of the system is analyzed.

2. Mathematical Model

2.1. Modified Normal Contact Force Model

Many researchers have highlighted that the improvement of the contact force model is a fundamental concern to accurately predict the dynamic responses of multibody systems [38]. The normal contact force models including Hertz, H-C and L-N are known as point contact and are appropriate for contact cases with large clearances, small loads and high restitution coefficients. Due to the characteristics of the flap actuation system (small clearance and heavy load), the contact area between the journal and the bearing is large and cannot be simulated by the above three models. In line with this situation, Gonthier [39] developed a volumetric contact force model that is suitable for large contact areas and applicable to a wide range of restitution coefficients. Moreover, the contact stiffness coefficient of the contact force model is not only related to the geometric properties, material of the collision body and clearance size, but also changes with the penetration depth and contact state [40]. Wang [41] presented a model with a variable contact stiffness

coefficient and bearing axial length to suit these conditions. Thus, based on the above two models, a modified normal contact force model is proposed in Reference [42].

$$F_n = \frac{\pi}{2} L \delta^n \left(\frac{1 - \nu_B^2}{E_B} + \frac{1 - \nu_J^2}{E_J} \right)^{-1} \left(\frac{1}{2(\Delta R + \delta)} \right)^{0.5} \left[1 + \frac{1 - c_r^2}{c_r} \frac{\dot{\delta}}{\dot{\delta}^{(-)}} \right] \quad (1)$$

where F_n is the normal contact force, L is the bearing's axial length, δ is the penetration depth, ΔR is the radial clearance, c_r is the restitution coefficient, $\dot{\delta}^{(-)}$ is the journal's initial velocity, and n is equal to 1.5 for metallic contact. ν_B , ν_J , E_B and E_J are Poisson's ratios and Young's moduli, and the symbols B and J are for the bearing and journal, respectively.

2.2. Tangential Contact Force Model

The Ambrósio model [43] is adopted to describe the friction phenomenon, which can be given as

$$F_t = -c_f c_d F_n \text{sgn}(v_t) \quad (2)$$

where v_t is the relative tangential velocity between the journal and bearing, c_f is the friction coefficient, and c_d is the dynamic correction coefficient, which can be expressed as

$$c_d = \begin{cases} 0 & v_t \leq v_0 \\ \frac{v_t - v_0}{v_1 - v_0} & v_0 \leq v_t \leq v_1 \\ 1 & v_t \geq v_1 \end{cases} \quad (3)$$

where v_0 and v_1 are the given tolerances for the relative tangential velocity.

2.3. Dynamics Equations of the Flap Actuation System

Defining that a multibody system includes m flexible components, the generalized coordinate vector q of the component i in the system is determined by its centroid coordinates R_i , azimuth Euler angle coordinates P_i and the modal coordinates τ_i , which can be expressed as

$$q_i = [R_i^T \quad p_i^T \quad \tau_i^T]^T \quad (4)$$

$$q = [q_1^T \quad q_2^T \quad \cdots \quad q_m^T]^T \quad (5)$$

Assuming that m components are connected by n joints, the motion constraint equation is expressed by the generalized coordinate vector as

$$\Phi^k(q, t) = [\Phi_1^k(q, t) \quad \Phi_2^k(q, t) \quad \cdots \quad \Phi_n^k(q, t)] \quad (6)$$

In order to ensure that the system has a definite motion, it is necessary to impose a driving constraint on the system so that the degree of freedom is zero. The driving constraint equation is represented as

$$\Phi^d(q, t) = 0 \quad (7)$$

Combined with Equations (6) and (7), the constraint equation is given as

$$\Phi(q, t) = [\Phi^k(q, t) \quad \Phi^d(q, t)]^T = 0 \quad (8)$$

When the joint clearance is not considered, according to the Lagrange multiplier method, the dynamics equation is expressed as [17]

$$\begin{cases} M\ddot{q} + C\dot{q} + Kq + \Phi_q^T \lambda = Q(q, \dot{q}) \\ \Phi(q, t) = 0 \end{cases} \quad (9)$$

where q , M , C , K and $Q(q, \dot{q})$ are the generalized matrices of the coordinates, mass, damping, stiffness and force of the system, respectively, Φ_q is Jacobian matrix of the constrained equation and λ is the Lagrange multiplier.

When the joint clearance is considered, the contact force model is embedded into the dynamics equation and a unit step function is introduced as

$$\mu(\delta) = \begin{cases} 0 & \delta < 0 \\ 1 & \delta \geq 0 \end{cases} \quad (10)$$

where δ is the penetration depth.

The equation of the contact force model is given as

$$F_c = \mu(\delta)(F_n + F_t) \quad (11)$$

where F_c is the contact force, including the normal contact force F_n and tangential contact force F_t . The normal contact force F_n can be obtained by Equation (1) and the tangential contact force F_t is calculated by Equation (2).

In view of these, the dynamics equation of the system with the joint clearance is obtained as

$$\begin{cases} M\ddot{q} + C\dot{q} + Kq + \Phi_q^T \lambda = Q(q, \dot{q}) + F_c \\ \Phi(q, t) = 0 \end{cases} \quad (12)$$

When the control performance of the EMA is considered, the dynamics equations of the rigid–flexible flap actuation system with EMA and joint clearance can be given as

$$\begin{cases} M\ddot{q} + C\dot{q} + Kq + \Phi_q^T \lambda = Q(q, \dot{q}, \chi) + F_c \\ \Phi(q, t) = 0 \\ g(q, \dot{q}, \chi, \dot{\chi}) = 0 \end{cases} \quad (13)$$

where $g(q, \dot{q}, \chi, \dot{\chi})$ is the generalized matrix of EMA state equation and χ is the generalized matrix of the state variable.

3. Modeling of the Flap Actuation System

3.1. Rigid–Flexible Model

As displayed in Figure 2, the flap actuation system is formed by the EMA, actuating rod, auxiliary lever arm, auxiliary rod, lever arm, shaft, inertia and some bearing houses. For the parameters of the materials for these parts please refer to Reference [42]. The force and torque output by the EMA make the actuating rod produce linear displacement and drive the linkage mechanism composed of the auxiliary lever arm, auxiliary rod and lever arm to rotate. The shaft is fixed with the lever arm, so the shaft deflection can be realized owing to the rotation of the lever arm.

As shown in Figure 2, four revolute joints are regarded as imperfect ones. The collision process of clearance joints is described by the modified normal contact force model and Ambrósio friction model. The contact force models are embedded into Adams environment through the user-defined sub-routine.

The linkage mechanism can be modelled to be flexible. The finite element models of the above three components are established by using ANSYS 18.0 and are generated into modal neutral files. Then, the files are imported into the rigid model under ADAMS environment and connected with other rigid components of the system through rigid nodes. Finally, the rigid–flexible coupling model of the system is obtained.

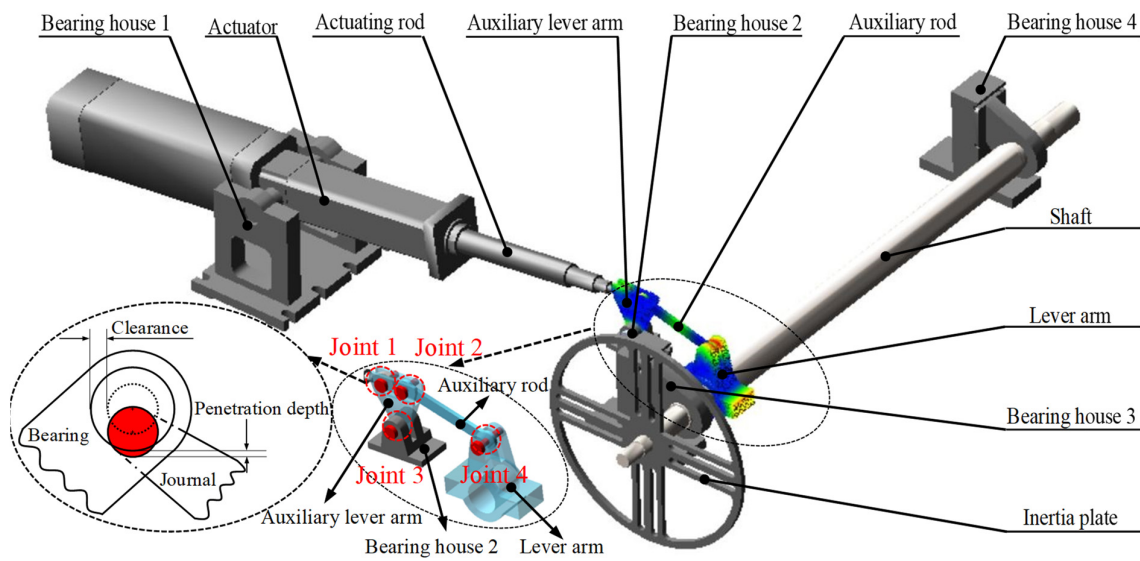


Figure 2. The rigid-flexible model of the flap actuation system.

The radial clearance size is equivalent to 0.1 mm with a bearing diameter of 15 mm [44]. And the bearing's axial length is equal to 15 mm. The friction coefficient and restitution coefficient are defined as 0.01 and 0.46 [45]. The given tolerances for the tangential velocity, v_0 and v_1 , are 0.1 and 1 mm/s [46,47]. The units of the angle, angular velocity and angular acceleration of the shaft are defined as deg, deg/s and deg/s², respectively. The relationship between “deg” and “rad” is “1 rad = 180 deg/π”.

3.2. Model of the Electromechanical Actuator (EMA)

As displayed in Figure 3, the simulation model of the EMA consists of four modules, including the control module, electrical module, mechanical module and interface module.

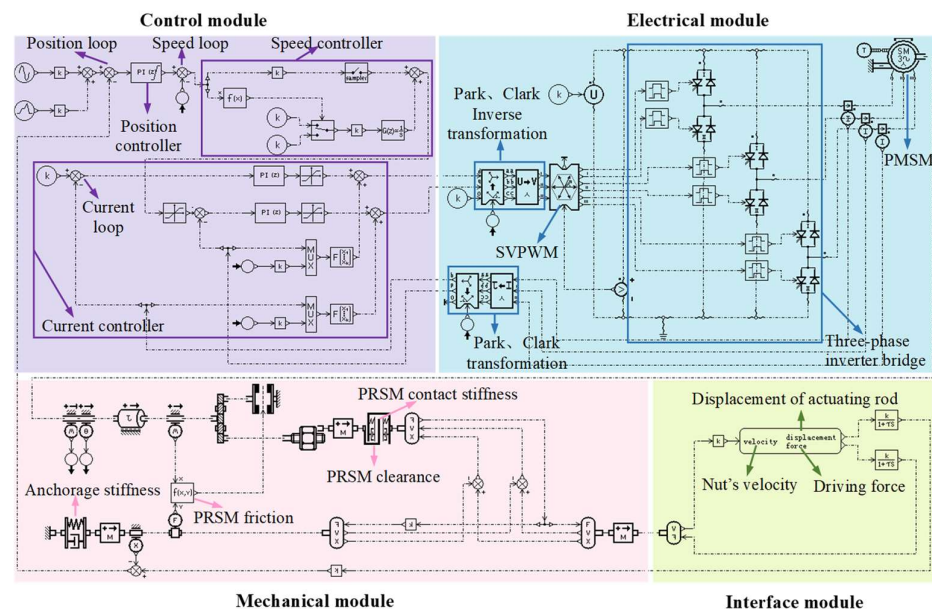


Figure 3. The EMA model with the “ $i_d = 0$ ” current vector control.

The EMA parameters are given in Table 1. The direct-drive EMA based on the planetary roller screw mechanism (PRSM) is studied, in which the PRSM is compacted with a motor rotor without a reducer. The direct-drive EMA has the characteristics of compact structure and high integration, which can reduce weight and volume [48].

Table 1. EMA Parameters.

Parameter	Value	Unit
Winding resistance	0.187	Ω
Winding inductance	4.07	mH
Back EMF constant	174	V/krpm
Electromagnetic torque constant	2.87	N·m/A
Rotor inertia moment	0.015	kg·m ²
Rated frequency	113	Hz
Rated voltage	370	V
Screw lead	8	mm
Nominal thrust	104	kN
Command of EMA displacement	10	mm
Rated torque	165	N·m
Stall torque	171	N·m
Rated speed	1800	rpm
Rated current	59.6	A
Stall current	65.7	A
Rated power	31	kW
Pole number of electrode	8	/
Effective stroke	250	mm
Nominal speed	240	mm/s
Motor friction coefficient	0.002	N·m/rpm
Command frequency	2	Hz

3.2.1. Electrical and Control Modules

The permanent magnet synchronous motor (PMSM) can be employed as the power source of an EMA, which is widely used in aerospace, industrial robots, electric vehicles and other fields with high control performance and precision because of its advantages of a high power factor, high efficiency, high reliability and small size [49,50].

In order to simplify the mathematical modeling process of the PMSM, the following assumptions are made: The effects of stator core saturation, eddy currents and hysteresis are ignored. The three-phase winding of the stator is uniformly and symmetrically distributed. The leakage inductance of the motor winding is neglected. The conductivity of the permanent magnet material is zero. The magnetic field generated by the winding exhibits a sinusoidal distribution [51].

Based on the above assumptions, the voltage equation of a PMSM in the d - q coordinate system is given as

$$\begin{cases} u_d = Ri_d + \frac{d}{dt}\psi_d - \omega_e\psi_q \\ u_q = Ri_q + \frac{d}{dt}\psi_q + \omega_e\psi_d \end{cases} \quad (14)$$

where u_d , u_q , i_d , i_q , ψ_d and ψ_q are the voltage, current and flux linkage of the stator in the d axis and q axis, respectively, R is the stator resistance and ω_e is the electric angular velocity.

The equation of the flux linkage is described as

$$\begin{cases} \psi_d = L_d i_d + \psi_f \\ \psi_q = L_q i_q \end{cases} \quad (15)$$

where L_d and L_q are the inductance of the stator in the d axis and q axis, respectively, and ψ_f is the flux linkage of the permanent magnet.

The voltage equation of the stator can be expressed as

$$\begin{cases} u_d = Ri_d + L_d \frac{d}{dt}i_d - \omega_e L_q i_q \\ u_q = Ri_q + L_q \frac{d}{dt}i_q + \omega_e (L_d i_d + \psi_f) \end{cases} \quad (16)$$

Coordinate transformation is utilized for simplifying the mathematical model of the PMSM. The equation of PMSM electromagnetic torque is written as

$$T_e = \frac{3}{2} p_n (\psi_d i_q - \psi_q i_d) = \frac{3}{2} p_n i_q [\psi_f + i_d (L_d - L_q)] \quad (17)$$

where T_e is the electromagnetic torque and p_n is the pole number of the three-phase PMSM.

The vector control method is always adopted into the PMSM for speed regulation and can improve the torque response and dynamic performance of the system. The “ $i_d = 0$ ” vector control method of PMSM is applied, so Equation (17) can be simplified as

$$T_e = \frac{3}{2} p_n \psi_f i_q \quad (18)$$

The EMA model with the “ $i_d = 0$ ” current vector control is represented in Figure 3, and a three-closed-loop control is performed. The control process is described as follows: when the position command is input, the speed control command is generated by the position controller and the reference current i_q is output by the speed controller. The input signal of the space vector pulse width modulation (SVPWM) is obtained by the inverse transformations of Park and Clark.

The SVPWM can calculate the motor control voltage and then output to the three-phase inverter bridge for providing the motor winding to drive the motor rotor. During the control process, the speed feedback of the motor rotor is provided by its position sensor and the current feedback is obtained by the three-phase current sensor. i_d and i_q are obtained through Park transformation and Clark transformation and are fed back to the current controller. The position feedback is provided by the linear displacement sensor. Finally, the three-closed-loop servo control of the EMA is realized.

3.2.2. Mechanical Module

The mechanical module is the planetary roller screw mechanism (PRSM), which can convert the rotational motion of the motor shaft into the linear displacement of the nut. The PRSM has become a new type of an actuator due to it being a small size, lightweight and pollution free and having a high bearing capacity and long life [49,52,53]. The effects of the contact stiffness and clearance of the PRSM as well as the EMA anchorage stiffness installed on the wing are considered in the EMA model, as shown in Figure 3.

Friction is one of the important characteristics of the planetary roller screw mechanism (PRSM). Excessive friction can lead to excessive wear between the meshing surfaces and temperature increases for various components in the PRSM, affecting its transmission accuracy, efficiency and dynamic performance. Equation (19) is proposed to describe the friction behavior of the PRSM. The Coulomb friction (Part 1), Stribeck effect at low speeds (Part 2) and Coulomb effect related to load and power quadrants (Part 3) are considered in Equation (19).

$$f = [f_c + f_s * e^{-|\omega|/\omega_{1r}} + |F_e|(b + c * \text{sgn}(\omega * F_e))] \text{sgn}(\omega) \quad (19)$$

where f_c is the Coulomb friction force, f_s is the Stribeck friction force, ω_{1r} is the constant that determines exponential decay, F_e is the load force, b is the average coefficient of the external force and c is the quadrant determination coefficient.

3.2.3. Interface Module

The interface module between the EMA model and the dynamic rigid–flexible model is modeled. In the interface module, the nut’s velocity is output from the EMA model (AMESim) to the dynamic flap actuation system (ADAMS). At the same time, the displacement of the actuating rod input from the ADAMS model to the EMA model in AMESim is achieved and the position closed-loop control is accomplished. The driving force calculated by the speed of the actuating rod is input from ADAMS to AMESim.

For the interface definition of the dynamics model in ADAMS, the displacement and force of the actuating rod as the data state variables, namely, ARRAY_1, ARRAY_2 and ARRAY_3 are defined as “displacement, force”, “time state variable” and “output state variables”, respectively. The nut’s velocity is defined as the input variable for parameter assignment, that is, the motion drive function is ARYVAL (ARRAY_3, 1) with the type of speed drive. The GSE state function is established to accomplish the data interaction.

4. Experiments and Model Verification

Based on the test rig of the flap actuation system, as shown in Figure 4, experiments with different frequencies of the sine-input control instruction and loads are performed. The experiments are conducted to verify the effectiveness of the co-simulation model. The diameters of four revolute joints are measured for three times and the average values are calculated, as listed in Table 2. The clearance values can be calculated and taken as the corresponding parameters for the simulation model.

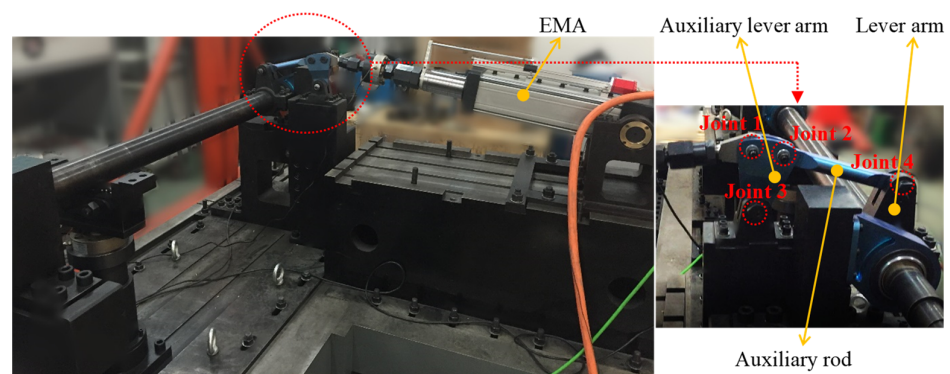


Figure 4. Test rig of the flap actuation system.

Table 2. Measured values of four revolute joints (Unit: mm).

Measured Values	Joint 1	Joint 2	Joint 3	Joint 4
Diameter of journals	14.957	14.959	14.955	14.958
Diameter of bearings	15.036	15.036	15.030	15.035

The sensors used for the measurements of the flap actuation system are listed in Table 3. The experimental procedure and method are described as follows. The amplitude of the sine signal for the EMA displacement is 10 mm and the frequency is 0.2–6 Hz (increasing by 0.2 Hz at 0.2–2 Hz and 0.5 Hz at 2–6 Hz, with 3 cycles as a group). The signal of the displacement sensor for the EMA and the encoder signal of the angular displacement for shaft are read. When the flap actuation system is loaded, the aerodynamic loads are simulated using a torque loading device.

Table 3. The sensors used for the measurements.

Sensor	Type	Position	Measurement
Torque sensor	Lorenz	Loading output end	Loading torque
Tension and pressure sensor	Interface	EMA output end	EMA output force
Incremental rotary encoder	Heidenhain-Rod426	Loading output end	Shaft’s angle
Grating ruler	Heidenhain	EMA output end	EMA output displacement

When the load is not applied, three different frequencies of sine-input instruction are defined, 1, 2 and 4 Hz. The simulations and experiments are compared, as displayed in Figures 5–7. The results show that: (1) For the simulation curves and the experimental

curves, because of nonlinear factors in EMA, amplitudes of the EMA displacement tend to decay with the increase in the command frequency. The higher the frequency, the more obvious the attenuation. (2) The simulation curves of shaft's angle tally well with the experimental ones. For a shaft's angular acceleration, the simulation and experimental curves show more obvious oscillation with pulse type than those for the shaft's angle. With the increase in command frequency, the oscillation is more obvious and the amplitude near the reversing position is higher.

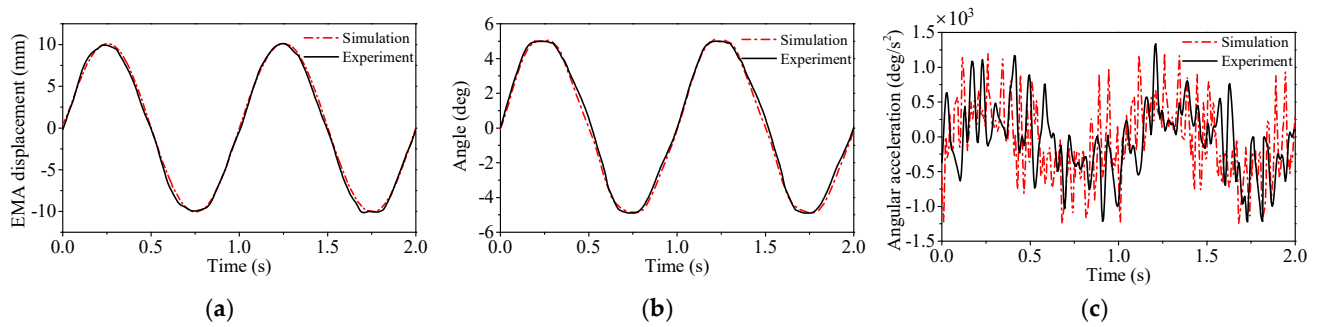


Figure 5. Comparison results with the EMA command frequency of 1 Hz and no load: (a) EMA displacement; (b) angle; (c) angular acceleration.

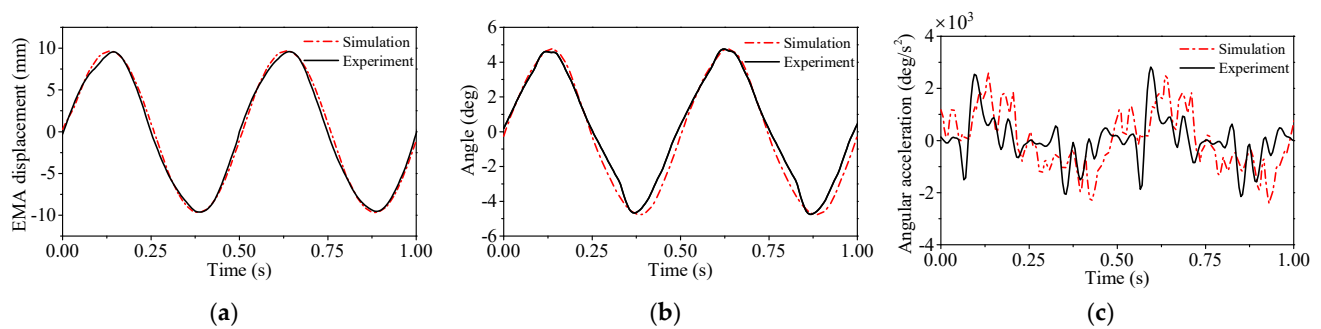


Figure 6. Comparison results with the EMA command frequency of 2 Hz and no load: (a) EMA displacement; (b) angle; (c) angular acceleration.

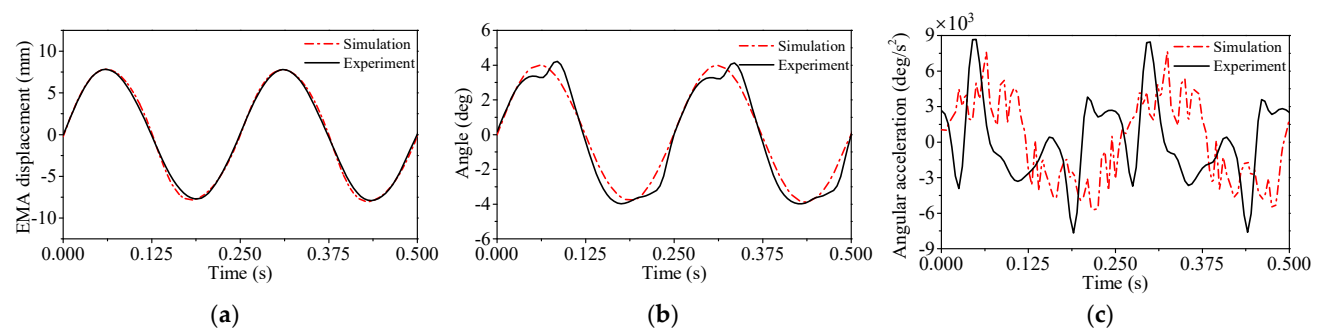


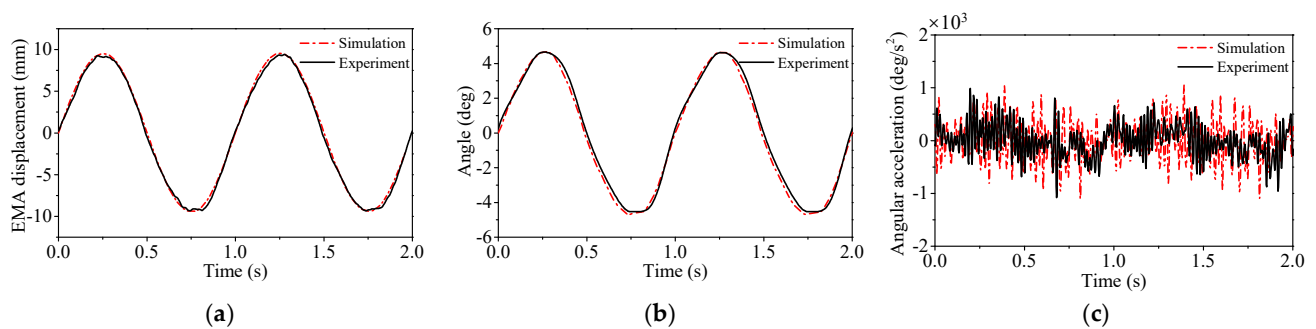
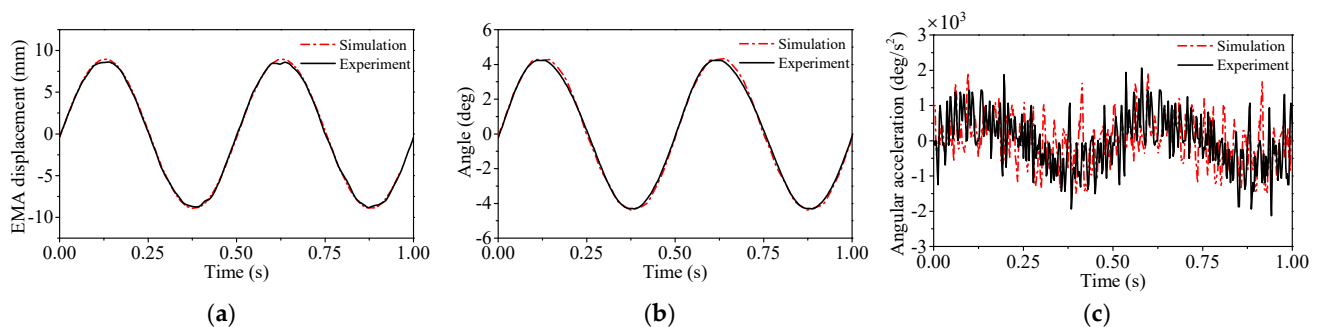
Figure 7. Comparison results with the EMA command frequency of 4 Hz and no load: (a) EMA displacement; (b) angle; (c) angular acceleration.

The comparison of response amplitudes at different frequencies without load is performed in Table 4. It can be found that with the increase in command frequency, peaks of EMA displacement and shaft's angle decrease, while that of angular acceleration increases. The reason is that when the flap actuation system moves at high frequencies, the EMA bandwidth is limited and the response is difficult to follow the expected command amplitude, which results in the amplitude reduction of the EMA displacement and shaft's angle. Besides, owing to the increase in the collision velocity of the clearance joint, the strong collision phenomenon makes the shaft's angular acceleration violent.

Table 4. Comparison with different frequencies under no-load conditions.

Response Amplitudes	Description	Frequency (Hz)		
		1	2	4
EMA displacement (mm)	Simulation	10.141	9.681	7.909
	Experiment	10.115	9.627	7.827
	Relative error	0.257%	0.561%	1.048%
Shaft's angle (deg)	Simulation	5.176	4.815	3.785
	Experiment	4.992	4.753	3.986
	Relative error	3.686%	1.304%	5.043%
Shaft's angular acceleration (deg/s ²)	Simulation	1207.843	2593.131	7703.624
	Experiment	1296.773	2819.805	8456.692
	Relative error	6.858%	8.039%	8.905%

With a torque load of 500 N·m, similarly, the EMA command frequency is selected as 1, 2 and 4 Hz, respectively. The comparison between the simulation and experiment is displayed in Figures 8–10. Different from the responses under the no-load conditions, as represented in Figure 11b, when the command frequency is 4 Hz, the experimental curve of the shaft's angle is distorted under the action of redundant torque. The effect of redundant torque is neglected in the loading process of the co-simulation model, so the simulation results are still approximate to sinusoidal curves, which leads to the deviation between the experiment curves.

**Figure 8.** Comparison results with the EMA command frequency of 1 Hz and load: (a) EMA displacement; (b) angle; (c) angular acceleration.**Figure 9.** Comparison results with the EMA command frequency of 2 Hz and load: (a) EMA displacement; (b) angle; (c) angular acceleration.

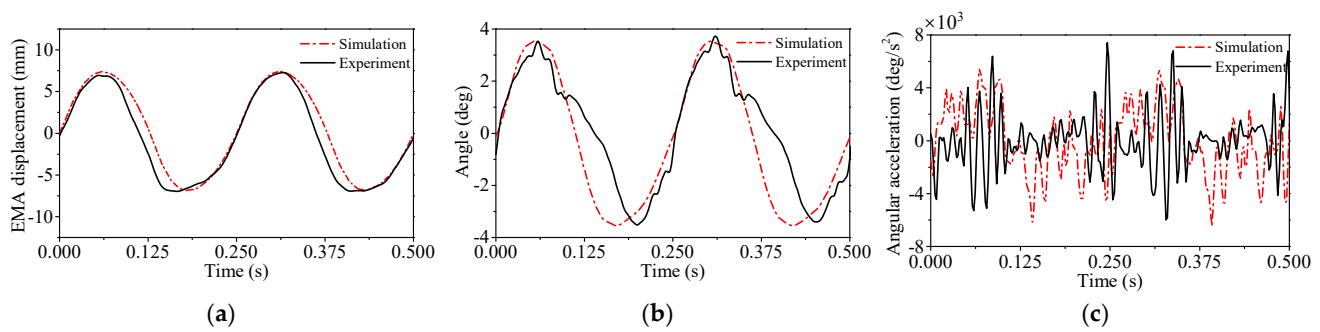


Figure 10. Comparison results with the EMA command frequency of 4 Hz and load: (a) EMA displacement; (b) angle; (c) angular acceleration.

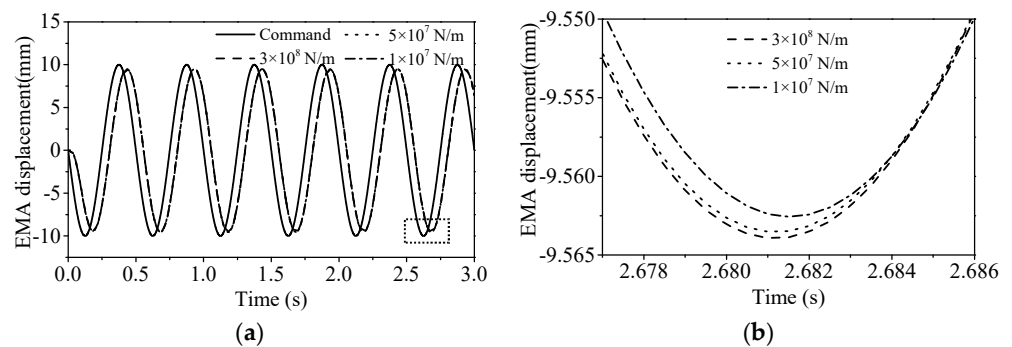


Figure 11. EMA displacement with different PRSM contact stiffnesses: (a) EMA displacement; (b) detail view.

The relative errors of the peaks between the simulation results and experimental results at different frequencies during loading are shown in Table 5. It can be found that the relative errors increase with the rise in command frequency. Compared with the system responses under no-load, the amplitudes of EMA displacement and the shaft's angle decrease during loading. The reason is that the EMA requires more thrust to overcome the load resistance when the torque load is applied. At this time, the EMA has to reduce the response amplitude in order to quickly follow the command. For the shaft's angular acceleration, its response frequency is higher and fluctuation amplitude is lower during loading than those with no-load, which means that the load can weaken the negative influence of joint clearance.

Table 5. Comparison with different frequencies under a load.

Response Amplitudes	Description	Frequency (Hz)		
		1	2	4
EMA displacement (mm)	Simulation	9.521	8.952	7.386
	Experiment	9.406	8.839	6.968
	Relative error	1.223%	1.278%	5.999%
Shaft's angle (deg)	Simulation	4.699	4.389	3.525
	Experiment	4.657	4.313	3.734
	Relative error	0.902%	1.762%	5.597%
Shaft's angular acceleration (deg/s ²)	Simulation	1054.038	1933.817	6448.451
	Experiment	1081.394	2125.249	7401.675
	Relative error	2.530%	9.008%	12.878%

From the comparison at different command frequencies under no-load and torque-load conditions, the results of the co-simulation model coincide well with the experimental

ones. The deviation between them might be caused by the following factors: (1) Parameters including the PRSM contact stiffness, PRSM clearance and EMA anchorage are regarded as certain values according to some references in the simulations, but those parameters are variables changing with the test conditions. (2) The joints between the bearing house 1 and EMA and between bearing houses 3 and 4 and the shaft are considered as ideal joints in simulations. However, the influence of the clearances existing in those joints is still inevitable due to processing and assembly, even if the influence of these clearances will be reduced and restrained as much as possible through lubrication. (3) The joint clearance size is invariant and the wear is not considered. In engineering, the joint clearance is irregular owing to the wear effect. (4) During the experiment, due to the limitation of hardware, the sampling rate of experimental data is low, which leads to a deviation of jitter degree.

As can be observed from Tables 4 and 5, the errors including the EMA displacement, shaft's angle and shaft's angular acceleration show an increasing trend with the increase in the command frequency, which is consistent with the conclusions in the literature [54], except the condition of shaft's angle without load at 2 Hz. The reason for this accident is due to a lot of uncertainties in the experiment and the randomness of experimental data.

In theoretical analysis, it is assumed that there is an in-plane positive collision between the journal and the bearing. In practice, there may be spatial contact between the two due to machining, manufacturing and assembly errors. In addition, the material restitution coefficient generally should be obtained through experimental testing. Under different heat treatment methods, the restitution coefficients are different. The restitution coefficient is defined as 0.46 in simulations [45], but it would be a variable in experiments.

5. Results and Discussion

5.1. Influence of PRSM Contact Stiffness

The PRSM contact stiffness can be regarded as meshing stiffness between the screw and the rollers or between the nut and the rollers. The PRSM stiffness is composed of three elements in series, including the cylinder stiffness, the contact stiffness and the thread stiffness. The magnitude of series stiffness is mainly determined by the weakest link in the series, and the contact stiffness is the weakest link. Contact stiffness has a significant impact on the overall stiffness, which directly affects the transmission efficiency and transmission performance of the PRSM. The EMA anchorage stiffness refers to the stiffness of the EMA installed on the flap. When the EMA anchorage stiffness is too low, it may lead to system flutter, which is detrimental to the flight control effect and aircraft structure.

The EMA anchorage stiffness is defined to be 1×10^8 N/m and the influence of PRSM clearance is not considered. Only the clearance at joint 1 is considered and is equal to 0.1 mm. The PRSM contact stiffness is selected as 3×10^8 , 5×10^7 and 1×10^7 N/m, respectively, and the corresponding responses of the EMA displacement, shaft's angle, angular velocity and angular acceleration with different PRSM contact stiffnesses are shown in Figures 11–14.

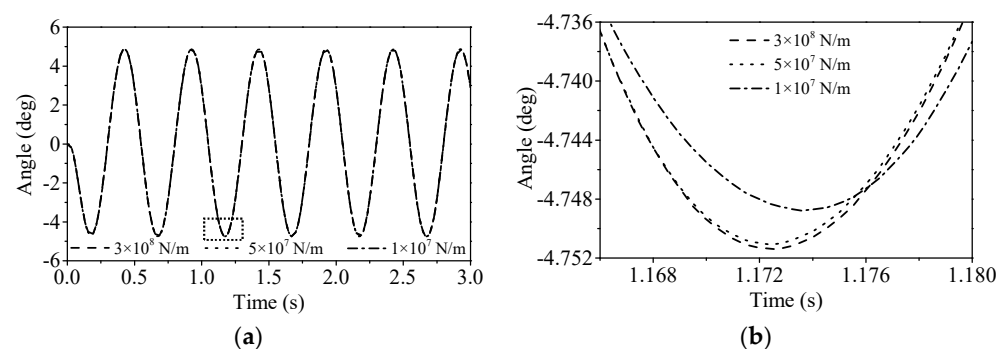


Figure 12. Angle with different PRSM contact stiffnesses: (a) angle; (b) detail view.

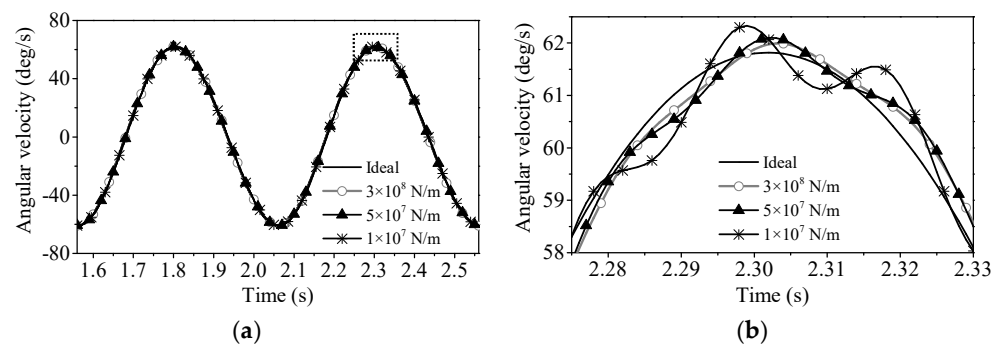


Figure 13. Angular velocity with different PRSM contact stiffnesses: (a) angular velocity; (b) detail view.

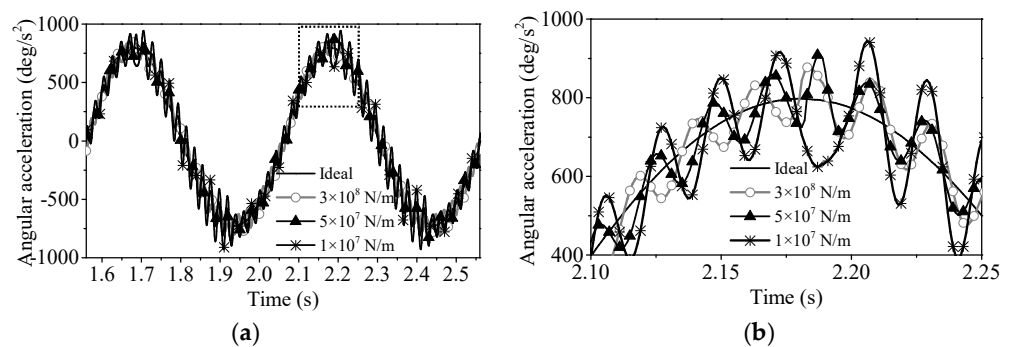


Figure 14. Angular acceleration with different PRSM contact stiffnesses: (a) angular acceleration; (b) detail view.

As represented in Figure 11, due to the influence of the EMA bandwidth, the EMA displacement at 2 Hz cannot follow and perfectly reproduce the command requirements in real time, and there is an amplitude attenuation and time lag. It can be seen from Figures 11 and 12 that EMA displacement and the shaft's angle are less affected by the PRSM contact stiffness. The maximum amplitudes of EMA displacement with different PRSM contact stiffnesses of 3×10^8 , 5×10^7 and 1×10^7 N/m are 9.5639, 9.5635 and 9.5626 mm, respectively, and those of the shaft's angle are 4.7514° , 4.7511° and 4.7488° . The attenuation degree of the response amplitude increases when the PRSM contact stiffness decreases.

Figure 13b shows that when the stiffness is equal to 3×10^8 N/m or 5×10^7 N/m, the curves of the shaft's angular velocity oscillate slightly near the ideal, but when the stiffness is 1×10^7 N/m, obvious fluctuations occur. The peak values of angular velocity fluctuations with the three kinds of stiffness are 61.893 deg/s, 62.101 deg/s and 62.328 deg/s, respectively, and the corresponding relative errors with the ideal one are 0.125%, 0.461% and 0.828%.

In Figure 14, the curves of angular acceleration represent high-frequency oscillations, and the smaller the stiffness, the more obvious the oscillation and the greater the fluctuation amplitude. In view of these, the PRSM contact stiffness has the greatest impact on the shaft's angular acceleration, the second is the angular velocity, and the smallest impacts are from the shaft's angle and the EMA displacement.

5.2. Influence of EMA Anchorage Stiffness

In this section, the PRSM contact stiffness is 1×10^8 N/m and the influence of the PRSM clearance is not considered. Only the clearance at joint 1 is considered as 0.1 mm. The EMA anchorage stiffness is selected as 3×10^8 , 5×10^7 and 1×10^7 , respectively, the responses with different EMA anchorage stiffnesses are shown in Figures 15 and 16.

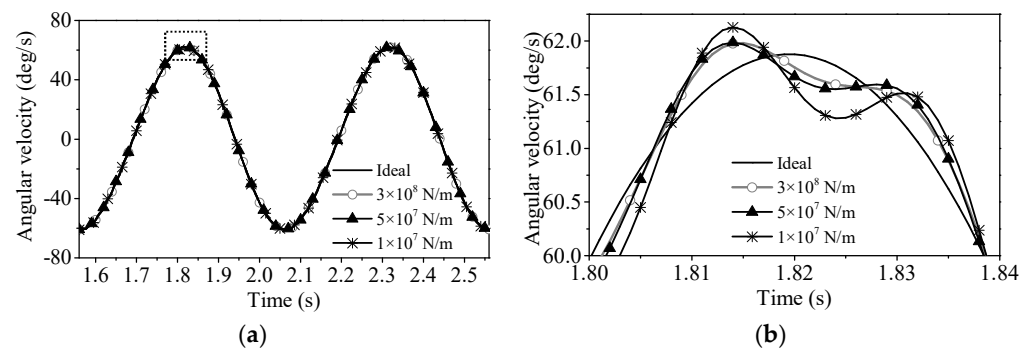


Figure 15. Angular velocity with different EMA anchorage stiffnesses: (a) angular velocity; (b) detail view.

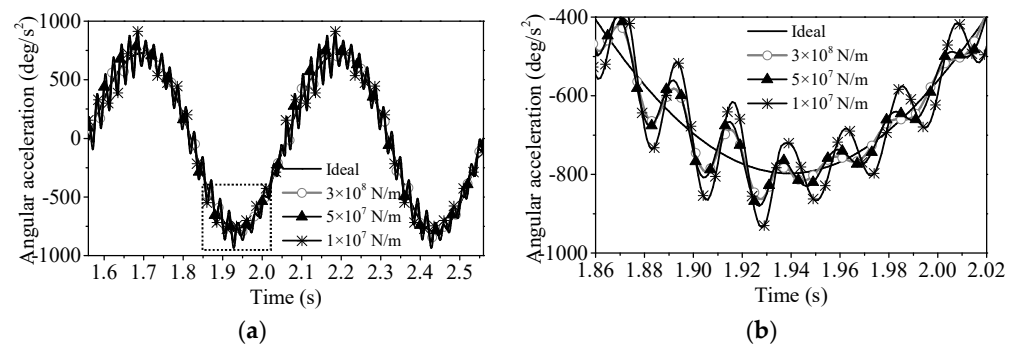


Figure 16. Angular acceleration with different EMA anchorage stiffnesses: (a) angular acceleration; (b) detail view.

The relative errors of the shaft's angular velocity with different EMA anchorage stiffnesses are 0.170%, 0.175% and 0.406%, respectively, and those of the shaft's angular acceleration are 10.291%, 11.350% and 16.823%. With the decrease in EMA anchorage stiffness, the oscillation degree and fluctuation amplitude of the dynamic responses increase significantly; the effect of the stiffness on the angular acceleration is evidently higher than that on the angular velocity.

5.3. Influence of PRSM Clearance

The PRSM clearance refers to the axial clearance between the screw and nut. The clearance can reflect the machining and installation accuracy of the PRSM. Clearance is one of the most important nonlinear influencing factors that limit the performance of the speed and position control. The position accuracy and stability of the EMA would be affected by the PRSM clearance.

The PRSM clearance is regarded as 0.5, 0.1 and 0.05 mm, respectively, and the responses with different PRSM clearances are given in Figures 17 and 18.

Compared with the stiffness influence, effects of the PRSM clearance on the angular velocity and angular acceleration are more obvious. Figure 17 shows that the clearance makes the angular velocity exhibit periodic oscillation and the fluctuation degree achieves the maximum near the system commutation. The approximate impulse responses in Figure 18 indicate that the fluctuation amplitude and oscillation frequency both increase as the clearance increases.

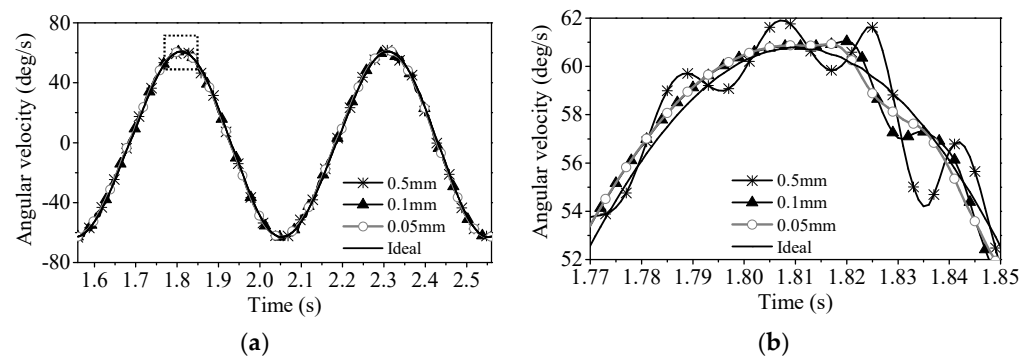


Figure 17. Angular velocity with different PRSM clearances: (a) angular velocity; (b) detail view.

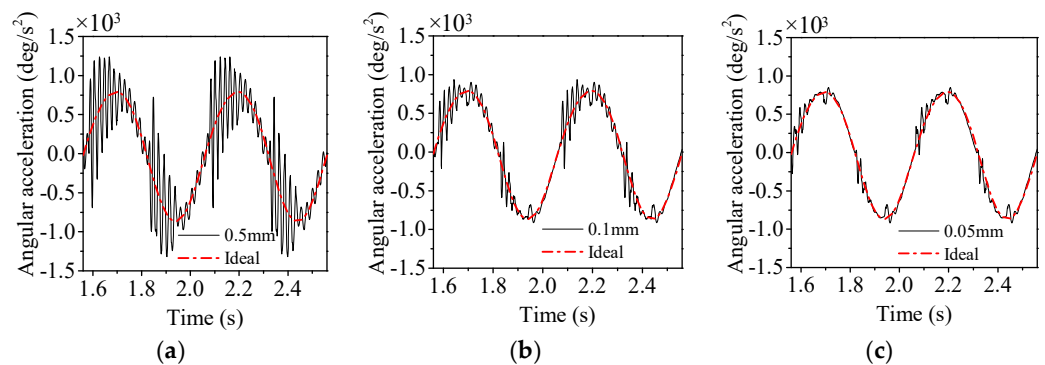


Figure 18. Angular acceleration with different PRSM clearances: (a) 0.5 mm; (b) 0.1 mm; (c) 0.05 mm.

As observed from Figures 14 and 16, when the PRSM contact stiffness or EMA anchorage stiffness is increased to 5×10^7 N/m, the difference in the system dynamic response is small compared to that with the stiffness of 3×10^8 N/m. To ensure the signal tracking ability of the EMA during the working process, in addition to adopting some control strategies, the stiffness should be appropriately increased in terms of structure. Therefore, it is recommended to design the PRSM contact stiffness or EMA anchorage stiffness to be at least 5×10^7 N/m.

Figures 17 and 18 show that the larger the PRSM clearance, the more significant the fluctuation amplitude of the system's response. In the EMA, the PRSM clearance not only affects control accuracy but also has a major impact on system stability. To improve the dynamic response of the EMA, a clearance eliminating structure is adopted mechanically. In terms of materials, wear-resistant materials are recommended to be used. Moreover, clearance compensation can be carried out through control strategies.

As mentioned above, nonlinear factors in the EMA such as the PRSM contact stiffness, EMA anchorage stiffness and PRSM clearance will affect the responses of the flap actuation system, especially the angular acceleration. Therefore, the EMA influence should be considered in the co-simulation model for analyzing the output characteristics of the aircraft.

5.4. Influence of the Number of Clearance Joints

The PRSM contact stiffness and EMA anchorage stiffness are both 3×10^8 N/m, regardless of the influence of the PRSM clearance. The clearances of the four joints are considered in turn and the clearance sizes are all defined to be 0.1 mm. The effect of flexible components is not considered. The number of clearance joints is selected as one (joint 1), two (joints 1, 2), three (joints 1, 2, 3) and four (joints 1, 2, 3, 4), respectively.

Figure 19 shows that when the number increased, the shaft's angle gradually fluctuates from the smooth curve and the deviation from the ideal value gradually increases. The shaft's angular velocity displayed in Figure 20 indicates that the responses with different numbers of clearance joints fluctuate around the ideal curve. The reason is that the contact

force acts on the system as a generalized external force. When there is only one clearance joint, the curve fluctuates smoothly, as shown in Figure 20a. Once multiple clearance joints are added, the responses are stepped. The interaction between multi clearance joints leads to the poor stability of the system.

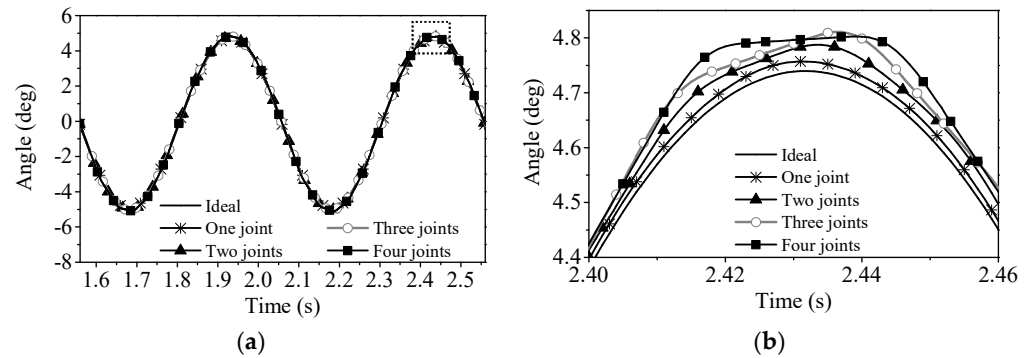


Figure 19. Shaft's angle with different numbers of clearance joints: (a) shaft's angle; (b) detail view.

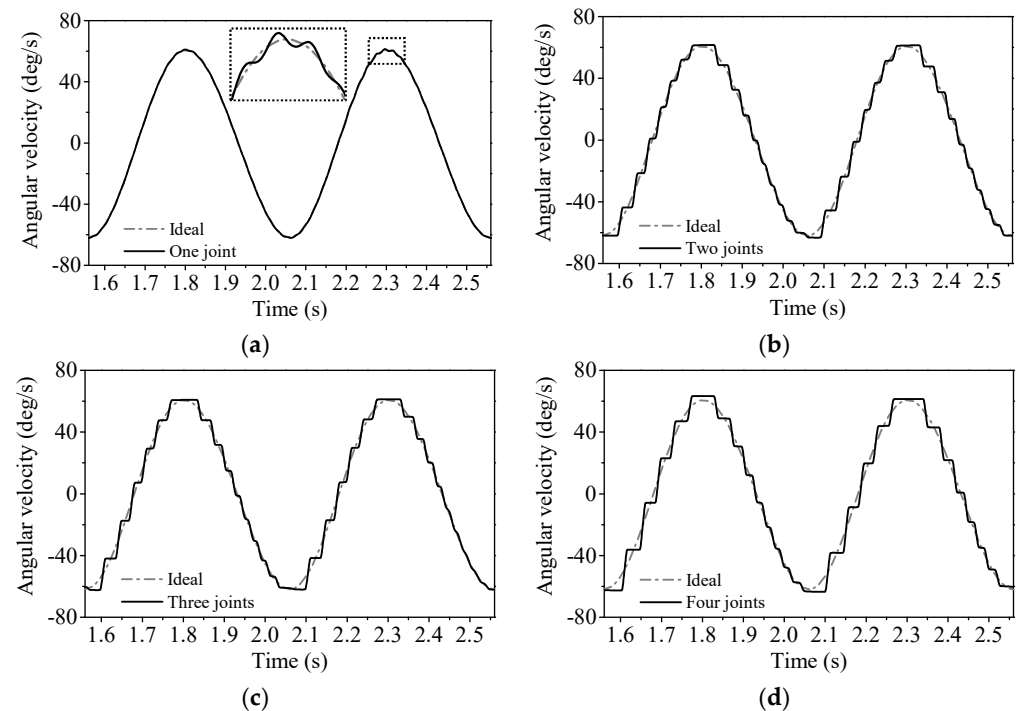


Figure 20. Shaft's angular velocity with different numbers of clearance joints: (a) one; (b) two; (c) three; (d) four.

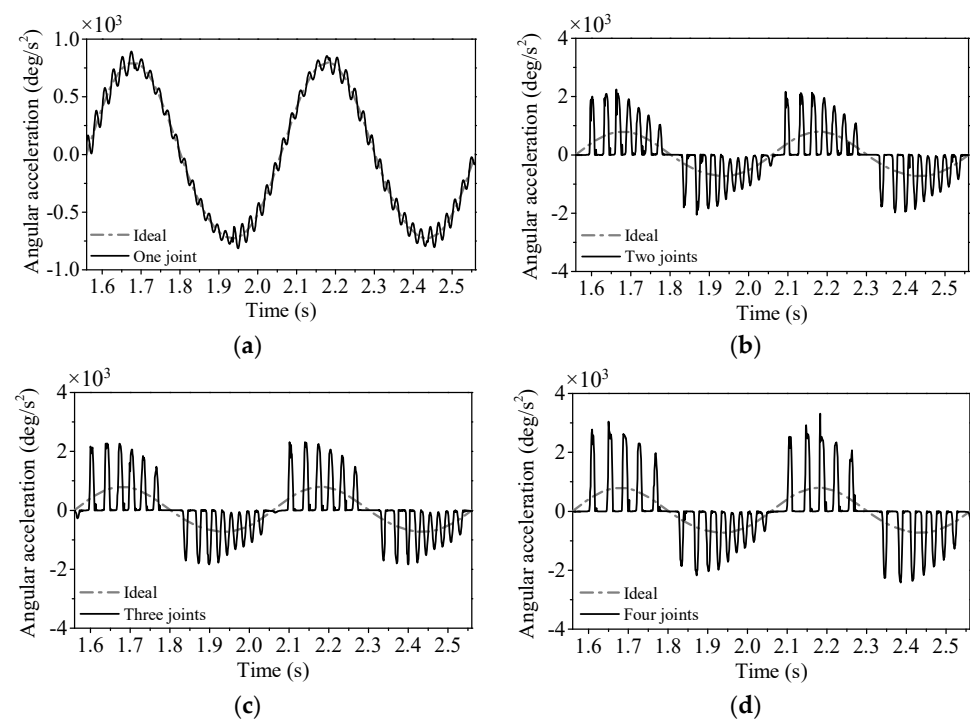
The fluctuation peaks and relative errors of shaft's angular velocity based on the dynamics model without the influence of EMA are compared with those of the co-simulation model in Table 6. The position drive of the EMA is defined as the ideal moving pair in the former model and the influence of nonlinear factors of the EMA is considered in the latter model.

Table 6. Fluctuation peak and relative error of the shaft's angular velocity with different numbers of clearance joints.

Models	Description	Number			
		1	2	3	4
Dynamic model without considering the influence of the EMA	Fluctuation peak (deg/s)	63.255	64.114	63.394	66.825
	Relative error	0.544%	1.909%	0.765%	6.218%
Co-simulation model	Fluctuation peak (deg/s)	61.505	63.251	62.028	65.580
	Relative error	1.173%	4.045%	2.033%	7.876%

The fluctuation peak is lower and the relative error is larger for the co-simulation model than the values for the dynamics model without the influence of the EMA in Table 6. The reason is that with the increase in the command frequency, the EMA displacement cannot reproduce the command and the amplitude shows attenuation. However, due to the influence of the stiffness, friction and other nonlinear links, the relative error based on the co-simulation model is high. Because of the fixed component in joint 3, the fluctuation peak and relative error of angular velocity with three clearance joints are less than those with two clearance joints. However, when four clearance joints are considered, owing to the interplay among multi clearance joints and the accumulated contact forces, the oscillation amplitude achieves the maximum value and the system response is more deviated from the ideal.

The angular acceleration curves based on the co-simulation model with different numbers of clearance joints are given. As displayed in Figure 21, the angular acceleration with one clearance joint oscillates smoothly around the ideal, while the curves are pulse shaped with multiple clearance joints.

**Figure 21.** Shaft's angular acceleration with different numbers of clearance joints: (a) one; (b) two; (c) three; (d) four.

The root mean square (RMS) error index is proposed to quantitatively reflect the influence of joint clearances. The comparison is based on the dynamics model without the influence of the EMA, and the co-simulation model is given in Table 7. With the increase in the number of clearance joints, whether using the dynamic model without considering the influence of the EMA or the co-simulation model, the fluctuation peak of the angular acceleration and the RMS index show upward trends. In addition, under the working conditions with the same number, the peak and RMS error index of the former model are greater than and less than the latter, respectively, which is consistent with the influence law of the number of clearance joints on the angular velocity shown in Table 6.

Table 7. Comparison results with different numbers of clearance joints.

Models	Description	Number			
		1	2	3	4
Dynamic model without considering the influence of EMA	Fluctuation peak (deg/s ²)	902.487	2482.807	2695.048	4063.383
	RMS error index	9.4225%	106.4543%	129.12%	152.1356%
Co-simulation model	Fluctuation peak (deg/s ²)	871.136	2127.027	2245.753	3310.093
	RMS error index	10.575%	114.585%	136.463%	161.414%

5.5. Influence of Flexibility

Flexibility is another important nonlinear factor affecting the system responses. The PRSM contact stiffness and EMA anchorage stiffness are both defined as 3×10^8 N/m. The influence of the PRSM clearance is not considered. Only the clearance at joint 1 is considered and it is assumed as 0.1 mm. When the auxiliary rod is considered to be rigid and flexible, the comparison results are displayed in Figures 22 and 23. The results show that the oscillation degree of the angular velocity and angular acceleration are significantly reduced when considering the component flexibility. The flexible component can restrain the vibration generated by the joint clearance, so that the joint collision effect is buffered.

As observed from Table 8, whether the dynamics model without the influence of the EMA or the co-simulation model are used, when the influence of component flexibility is considered, the fluctuation peak and RMS error index of the shaft's angular acceleration are smaller than those of the rigid-body model. Besides, whether using the rigid model or the rigid-flexible coupling model, the RMS error index of the co-simulation model is higher than that of the dynamics model without the influence of the EMA. The reason is that the superposition of the nonlinear properties of the EMA, flexibility and joint clearance makes the dynamic response more volatile.

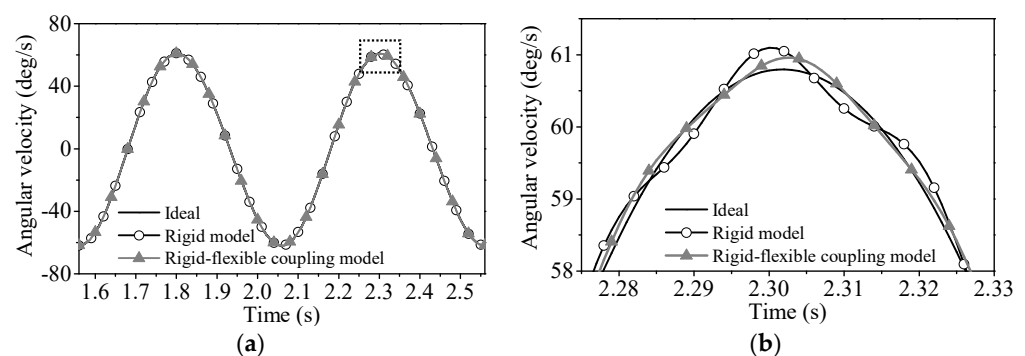


Figure 22. Shaft's angular velocity with the component flexibility: (a) shaft's velocity; (b) detail view.

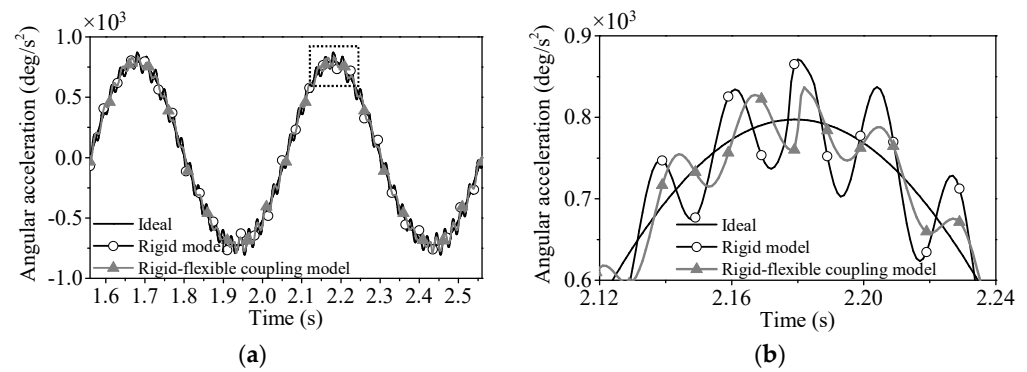


Figure 23. Shaft's angular acceleration with the component flexibility: (a) shaft's acceleration; (b) detail view.

Table 8. Comparison results with or without flexibility.

Models	Description	Influence of Flexibility	
		Rigid Model	Rigid-Flexible Coupling Model
Dynamic model without considering the influence of EMA	Fluctuation peak (deg/s ²)	902.487	877.249
	RMS error index	9.423%	5.485%
Co-simulation model	Fluctuation peak (deg/s ²)	871.136	837.153
	RMS error index	10.575%	8.387%

5.6. Influence of Load

The parameter definitions for the EMA in this section are consistent with those in the section on the influence of flexibility. Only the clearance at joint 1 is considered and is equal to 0.1 mm. The load acting on the shaft is selected as 500, 1000 and 2000 N·m, respectively. Figure 24b indicates that when the load is not considered, the shaft's angular velocity with joint clearance fluctuates up and down around the curve without clearance. When the load is applied, the oscillation phenomenon disappears and the response curve becomes smooth. The relative errors of the fluctuation peak of angular velocity compared with the ideal are 0.393%, 0.732% and 1.303%, respectively, that is, the greater the load, the larger the relative error.

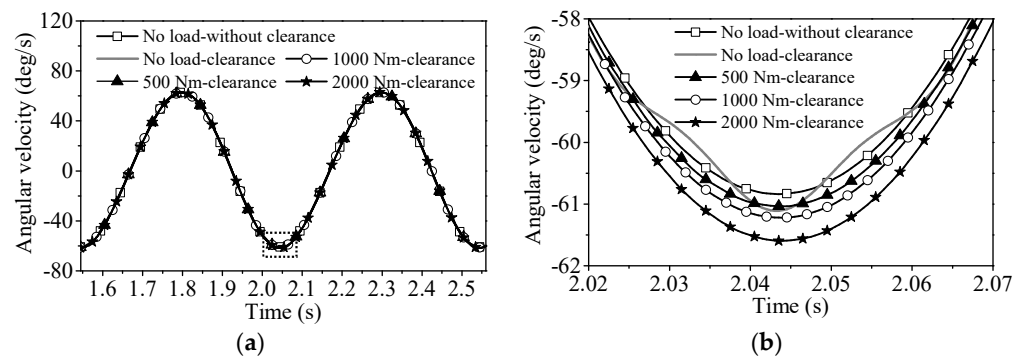


Figure 24. Shaft's angular velocity with different loads: (a) shaft's angular velocity; (b) detail view.

As shown in Figure 25, when the load is not considered, the joint clearance makes the angular acceleration oscillate sharply around the ideal one. When the load is added, the angular acceleration curve only oscillates when the system is reversing and the oscillation

gradually weakens under the effect of system damping until it becomes a smooth curve. When the joint clearance is considered, the RMS error index without load and with loads of 500, 1000 and 2000 N·m are 10.575%, 1.927%, 1.426% and 1.581%, respectively, which indicates that the load can absorb the energy and weaken the negative effects caused by the joint clearance. The system response becomes smooth and the stability is improved. However, the system accuracy is reduced with a too large load.

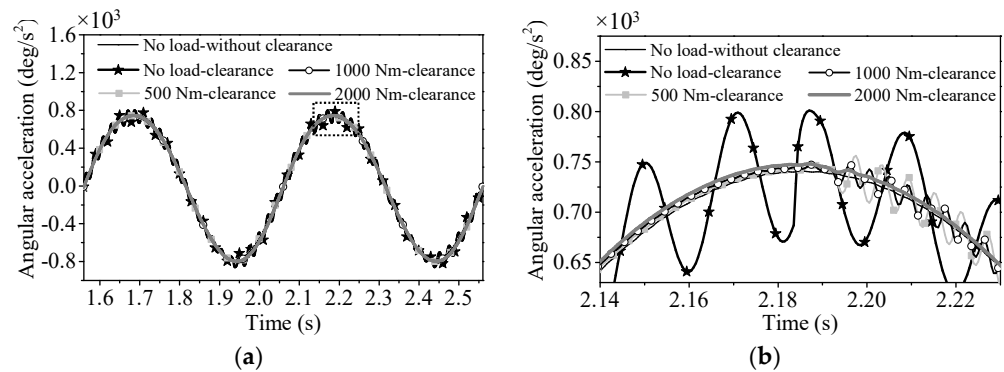


Figure 25. Shaft's angular acceleration with different loads: (a) shaft's angular acceleration; (b) detail view.

6. Conclusions

Based on the modeling and experimental verification, a co-simulation model of the flap actuation system is presented that takes the nonlinear properties of the transmission chain, including the electromechanical actuator (EMA) and the linkage mechanism, into account.

A dynamic rigid–flexible model is proposed, where the flexibility of the linkage mechanism and the clearances of four joints are considered. The contact force model is utilized to simulate the collision process of the journal and the bearing. Moreover, an EMA model composed of the control module, electrical module, mechanical module and the interface module is developed. The interaction variables are defined by setting the input and output interfaces between the EMA model and the dynamic rigid–flexible model to establish a co-simulation model. In addition, the experiments are conducted and the results agree well with the simulation ones.

On the grounds of the proposed co-simulation model of the system, the effects of the nonlinear properties including the contact stiffness of the planetary roller screw mechanism (PRSM), EMA anchorage stiffness, PRSM clearance, number of clearance joints, flexibility and load are investigated. The results indicate that: (1) The influence of the two types of stiffness is alike. The amplitude attenuation degree and the phase lag time of dynamic behaviors both increase with decreasing stiffness. (2) The PRSM clearance leads to the high-frequency pulse phenomenon of angular acceleration. (3) The interaction between clearance joints makes dynamic responses pulse. (4) The flexibility and load can effectively restrain the oscillation behaviors attracted by the joint clearance. Multiple clearance joints result in step responses with high amplitudes, and it is necessary to reduce the number of clearance joints through lubrication. In order to mitigate the adverse effects of clearances, appropriate component flexibility and loads can be changed to achieve this goal.

This system-level co-simulation method is also applicable to any servo system driven by an actuator, providing theoretical support for engineering applications.

It also should be noted that:

(1) The factors such as the geometric shape of the collision process, material characteristics, axial dimensions of the bearing and energy loss during the collision process are considered in the contact force model established in this paper. However, in the modeling and simulation process, it is assumed that the joint clearance value remains constant, which is a regular circular clearance. In engineering experiments, the clearance will not be a constant value, so it is necessary to consider the surface wear and incessant material loss

from the surface. Therefore, in subsequent research, the influence of wear on the surface profile of joint elements should be considered.

(2) The parameters such as inductance, resistance and rotor flux of the model of the permanent magnet synchronous motor (PMSM) established in this paper are assumed to be constant, without considering the losses produced by the copper, iron and eddy current. In practical engineering applications, the control effect of motors would be influenced by the above nonlinear factors. Therefore, in subsequent research, a more realistic PMSM model should be established. In addition, in terms of the control strategy, a traditional PID control algorithm is designed in this paper. Although the control effect of the PID algorithm has been verified through simulations to meet the requirements of the use of the EMA, in order to further improve the dynamic performance, it is necessary to design advanced control algorithms with learning, recognition, adaptation and fuzzy processing capabilities in the future research.

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Nomenclature

Symbols/acronyms	Meaning
F_n	Normal contact force
L	Bearing’s axial length
d	Penetration depth
ΔR	Radial clearance
c_r	Restitution coefficient
$\dot{\delta}^{(-)}$	Journal’s initial velocity
$n_{B,J}$	Poisson’s ratio of the bearing and journal
$E_{B,J}$	Young’s modulus of the bearing and journal
F_t	Tangential contact force
v_t	Relative tangential velocity
c_f	Friction coefficient
c_d	Dynamic correction coefficient
$v_{0,1}$	Given tolerances for the relative tangential velocity
q	Generalized matrix of the coordinates
M	Generalized matrix of the mass
C	Generalized matrix of the damping
K	Generalized matrix of the stiffness
$Q(q, \dot{q})$	Generalized matrix of the force
F_c	Contact force
$g(q, \dot{q}, \chi, \dot{\chi})$	Generalized matrix of EMA state equation
c	Generalized matrix of the state variable
$u_{d,q}$	Voltage of the stator in d axis and q axis, respectively
$i_{d,q}$	Current of the stator in d axis and q axis, respectively
$y_{d,q}$	Flux linkage of the stator in d axis and q axis, respectively

R	Stator resistance
ω_e	Electric angular velocity
L_d, L_q	Inductance of the stator in d axis and q axis, respectively
ψ_f	Flux linkage of the permanent magnet
T_e	Electromagnetic torque
p_n	Pole number of the three-phase PMSM
T_e	Electromagnetic torque
p_n	Pole number of the three-phase PMSM
$\psi_{d,q}$	Flux of the stator in d axis and q axis
$i_{d,q}$	Current of the stator in d axis and q axis
$L_{d,q}$	Inductance of the stator in d axis and q axis
ψ_f	Flux of the permanent magnet
f	Friction force of the PRSM
f_c	Coulomb friction force
f_s	Stribeck friction force
w_{1r}	Constant that determines exponential decay
F_e	Load force
b	Average coefficient of the external force
c	Quadrant determination coefficient
EMA	Electromechanical actuator
PMSM	Permanent magnet synchronous motor
SVPWM	Space vector pulse width modulation
PRSM	Planetary roller screw mechanism
RMS	Root mean square

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