MODELING AND ANALYSIS OF ELECTRO-HYDROSTATIC ACTUATORS EFFICIENCY

Chaofan Jiang[1,2,3,4], Yaoxing Shang[1,2,3,4], Tian Yu[1,2,3,4,*] Zhicheng Liu[1,2,3,4], & Mingyang Li[2,3]

[1]School of Automation Science and Electrical Engineering, Beihang University, Beijing, China.
[2]National Key Laboratory of Aircraft Integrated Flight Control, Beihang University, Beijing, China
[3]Ningbo Institute of Technology, Beihang University, Ningbo, China.
[4]Tianmushan Laboratory, Xixi Octagon City, Yuhang District, Hangzhou,China.

Abstract

Electro-hydrostatic Actuator (EHA) is an important part which is used to control the control surface of aircraft. EHA has the advantage of higher efficiency compared with traditional hydraulic system. Compared with the hydraulic actuator, EHA adds motor, hydraulic pump, booster tank and other components. Therefore, the calculation of EHA efficiency is also more complex than that of hydraulic actuators. After research and analysis, the efficiency of wet motor-pump is low under large load or high speed. Using a passive load-sensitive EHA configuration can improve this situation. The motor copper loss can be reduced by reducing the displacement of piston pump under large load. But the displacement cannot be reduced blindly, which will lead to dynamic decline. The influence of displacement change velocity on dynamic and energy loss was studied. The effects of load sensitive variable displacement mechanism parameters such as piston area, spring stiffness, radius of damping hole and initial pressure on the velocity of displacement change were studied. The influence of parameters on dynamics and energy loss is obtained. So in order to balance efficiency and dynamics, particle swarm optimization method is adopted. Piston area, spring stiffness, radius of damping hole and initial pressure of variable displacement are taken as optimization variables. Energy loss and step response risetime are taken as optimization objectives. The energy loss and risetime are kept at a low level by obtaining appropriate load sensing mechanism parameters. The results show that the passive load sensing mechanism parameters obtained by particle swarm optimization algorithm can reduce the energy consumption of EHA under large load conditions and improve the efficiency of EHA without reducing the dynamic of EHA.

Keywords: Electro-hydrostatic Actuators, Passive load-sensitive, Efficiency

1. Introduction

Electro-hydrostatic actuator (EHA) is an important form of power by wire. Compared with the traditional hydraulic actuation system has a higher efficiency. The efficiency of the actuator affects the power index of the aircraft energy system and the design of the actuator. Therefore, it is very important to improve the efficiency of the actuator. Compared with the hydraulic actuator, EHA adds motor, hydraulic pump, booster tank and other components. Therefore, the calculation of EHA efficiency is also more complex than that of hydraulic actuators. Depending on changes in motor speed and pump displacement, EHA can be divided into a variety of configurations. Including constant displacement pump and variable speed motor EHA, variable displacement pump and constant speed motor EHA, variable displacement pump and variable speed motor EHA [1-3]. EHA also has a variety of working conditions, including large load conditions. Under heavy load conditions, the copper loss is too large because the motor current is too high. There are the following studies on this situation. Jiao Z proposed direct load-sensitive EHA and active load-sensitive EHA to solve the problem of high energy consumption under large load conditions. Song Z [4] designed and analyzed direct load-sensitive EHA. It is concluded that direct load sensitive EHA can reduce motor torque under large load conditions. This reduces motor heating and improves EHA efficiency. But when the load changes from large to small, the pump swash plate recovery speed is slow. The displacement can not be recovered in time, so the system dynamic reduction. Li Z [5, 6] conducted research on active load sensitivity EHA. In order to improve the dynamic performance of the direct load-sensitive EHA, a special pressure control valve is added between the load-sensitive pressure and the load-sensitive variable displacement actuator. Thus, it can not only reduce the motor energy loss at large loads, but also meet the dynamic requirements at small loads. Li Y [7] took maximum displacement, minimum displacement, initial variable displacement pressure and moment of inertia as design variables. Energy loss and quality are taken as objective functions. The direct load sensitive EHA is optimized.

Based on previous studies, it is suggested that passive load sensitive EHA scheme can be used for large load conditions. The principle is shown in Fig. 1. Passive load sensitive EHA has a simpler structure than active load sensitive EHA. Compared to direct load sensitive EHA has better dynamics. The displacement of the pump will decrease with the increase of the load. Thus reduce motor current and improve efficiency. At the same time, when the load is reduced, the displacement can be restored through the pressure relief of the check valve. Thus to ensure the dynamic requirements of the actuator. A new variable displacement mechanism is added to the system. Includes hydraulic cylinder, return spring, damping hole, shuttle valve and check valve. The choice of these parameters affects the energy consumption and dynamics of the EHA. This paper mainly studies the effects of piston area, spring stiffness, damping hole size and initial pressure of displacement change on the velocity of displacement change. Then, the influence of displacement velocity on the energy loss and dynamics of the actuator is studied. Finally, particle swarm optimization algorithm is used to optimize the value of the variable. Compared with the simulation results, the passive load-sensitive EHA can reduce energy loss and improve efficiency under the premise of ensuring dynamic performance.

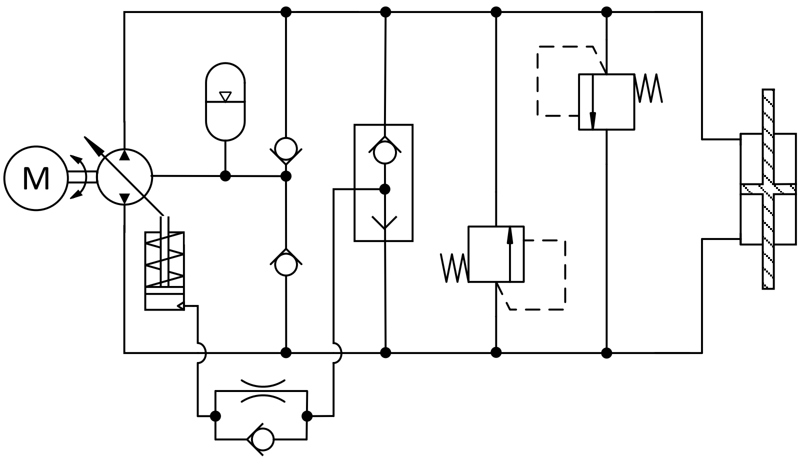

Figure 1 – Passive load sensitive EHA schematic.

2. Modeling of EHA

2.1 Passive Load Sensitive EHA Modeling

The main components of passive load-sensitive EHA are modeled: piston pump, motor, hydraulic cylinder, booster tank and load-sensitive module.

Flow equation of piston pump:

$$Q_{p1} = Q_{p2} = D_p \cdot \omega - K_{lp} \cdot \left(P_{p1} - P_{p2}\right) \tag{1}$$

Where Q_{p1} and Q_{p2} are the pump inlet and outlet flow, D_p is the displacement of the pump, ω is the angular velocity of the motor-pump, K_{lp} is the leakage coefficient of the pump, P_{p1} and P_{p2} are the inlet and outlet pressure of the pump.

Flow continuity equation of hydraulic cylinder is

$$\begin{aligned} Q_{c1} - K_{lc}\left(P_1 - P_2\right) &= \frac{V_{10} + AX_p}{B}\dot{P}_1 + A\dot{X}_p \\ K_{lc}\left(P_1 - P_2\right) - Q_{c2} &= \frac{V_{20} - AX_p}{B}\dot{P}_2 - A\dot{X}_p \end{aligned} \tag{2}$$

Where, Q_{c1} and Q_{c2} are the flow rate of two chambers of the hydraulic cylinder, K_{lc} is the leakage coefficient of the hydraulic cylinder, P_1 and P_2 are the pressure of two chambers of the hydraulic cylinder, V_{10} and V_{20} are the initial volume of two chambers of the hydraulic cylinder, A is the effective area of the piston of the hydraulic cylinder, X_p is the displacement of the piston rod, B is the bulk modulus of the hydraulic oil.

Piston dynamic balance equation is

$$\left(P_1 - P_2\right)\cdot A = m\ddot{X}_p + B_c\dot{X}_p + F_l \tag{3}$$

Where m is the mass of piston rod, B_c is damping coefficient of piston rod, F_l is the load force.

Voltage balance equation of motor is

$$U = Ri + L\dot{i} + E \tag{4}$$

Where U is the armature voltage of the motor, R is the resistance of the armature loop, i is the armature current, L is the inductance of the armature and E is the armature back electromotive force.

The equation of the back electromotive force is

$$E = K_e\omega \tag{5}$$

Where K_e is the back electromotive force coefficient of the motor.

The torque dynamic balance equation of the motor is

$$K_t i = J\dot{\omega} + B_m\omega + T_p \tag{6}$$

Where K_t is the torque coefficient of the motor, J is equivalent to the total moment of inertia of the motor-pump shaft, B_m is the total damping coefficient equivalent to the motor armature, T_p is the motor load torque, load torque drive pump work, $T_p = D_p(P_{p1}\text{-}P_{p2})$.

The relationship between the input flow Qa of the booster tank and the pressure Pa of the booster tank is as follows:

$$P_a = \frac{P_{ai}V_{gi}^k}{\left(V_{gi} - \int Q_a dt\right)^k} \tag{7}$$

Where P_{ai} is the initial pressure of the accumulator, V_{gi} is the initial volume of the gas in the accumulator, and k is the polytropic index of the gas.

Modeling load sensitive modules. The relationship between the hydraulic cylinder of the variable displacement mechanism and the swash plate is shown in the Fig. 2. Variable displacement mechanism hydraulic cylinder flow continuity equation:

$$Q_{vp} = \frac{V_{vp0} + A_{vp}X_p}{B}\dot{P}_r + A_{vp}\dot{X}_{vp} \tag{8}$$

In the formula, Q_{vp} is the flow rate of the hydraulic cylinder of the variable displacement mechanism, V_{vp0} is the initial volume of the hydraulic cylinder of the variable displacement mechanism, A_{vp} is the effective area of the piston, P_r is the pressure in the cylinder, and X_{vp} is the displacement of the piston.

Piston dynamic balance equation of variable displacement mechanism:

$$P_r\cdot A_{vp} - F_s = \left(\frac{J_s}{h^2} + m_{vp}\right)\ddot{X}_{vp} + B_c\dot{X}_{vp} + KX_{vp} \tag{9}$$

Where, F_s is the return spring preload, J_s is the swash plate moment of inertia, m_{vp} is the piston mass, h is the vertical distance between the piston and the swash plate line, and K is the return spring stiffness.

Damping hole flow formula:

$$Q_o = C_d A_o \sqrt{\frac{2\left|P_3 - P_r\right|}{\rho}} \tag{10}$$

Where Q_o is the flow through the damping hole, P_3 is the outlet pressure of the shuttle valve, ρ is the oil density, C_d is the flow coefficient, and A_o is the area of the small hole.

Relation between piston displacement of variable displacement mechanism and swash plate Angle:

$$X_{vp} = h\left(\beta - \alpha\right) \tag{11}$$

Where, β is the swash plate angle at the maximum displacement, α is the swash plate angle during the displacement change.

The relationship between swash plate Angle and displacement change:

$$D_p = D_{\max}\frac{\alpha}{\beta} \tag{12}$$

Where, D_{max} is the maximum displacement.

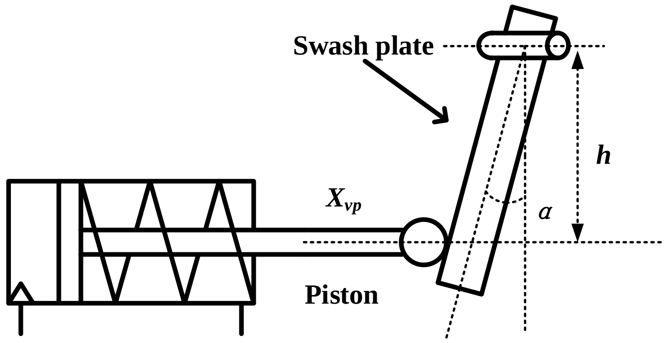

Figure 2 – Variable displacement mechanism hydraulic cylinder and swash plate.

2.2 EHA Energy loss Modeling

EHA energy loss mainly includes motor energy loss, piston pump energy loss and hydraulic cylinder energy loss.

Motor energy loss mainly includes electrical loss and mechanical loss. The electrical loss consists of copper loss and iron loss. The motor copper loss expression is:

$$P_{\mathrm{Cu}} = I^2 R \tag{13}$$

Where P_{cu} is the motor copper loss and I is the RMS phase current of the motor winding.

The motor iron loss expression is:

$$P_{\mathrm{Fe}} = k_{fe} f^{1.3} B_f^{\ 2} m_s \tag{14}$$

Where P_{fe} is the iron loss, k_{fe} is the stator material loss coefficient, f is the alternating magnetic field frequency, B_f is the maximum magnetic flux density, and m_s is the stator mass.

Because EHA adopts wet integrated motor-pump structure. Therefore, the motor cavity is filled with hydraulic oil, and the motor produces stirring losses. Therefore, the mechanical loss of the motor includes bearing friction loss and stirring loss. The bearing friction loss is small and negligible. Jin W [8] and Li Y [9] studied the stirring loss of wet motor, Li Y' s formula is:

$$P_{mf} = \begin{cases} 4\pi\mu\omega^2 L_{ef}\dfrac{R_{\mathrm{r}}^2 R_{\mathrm{s}}^2}{R_{\mathrm{s}}^2 - R_{\mathrm{r}}^2} \\ 2\pi R_r^{\ 2} L_{ef}\rho\cdot\varphi v\omega^2 \end{cases} \tag{15}$$

Where μ is the hydraulic oil viscosity, L_{ef} is the axial length of the stator and rotor, R_r is the rotor radius, R_s is the stator radius, and φ is turbulent coefficient which need to be experimental fitted, Li Y learned through the experiment that φ takes 210, v is kinematic viscosity of the oil.

The loss of the piston pump consists of volume loss and mechanical loss. The mechanical loss is mainly caused by the friction of the friction pair, the oil viscosity and the agitation loss. The formula is [7] :

$$P_{pf} = C_s p \omega D_p + C_v \mu \omega^2 D_p \tag{16}$$

Where C_s is the friction loss coefficient, p is the working pressure of the piston pump, C_v is the viscous friction coefficient.

The volume loss formula of the piston pump is [7] :

$$P_{pv} = C_{sv} \frac{D_p}{2\pi\mu} p^2 \tag{17}$$

In the formula, C_{sv} is laminar flow leakage coefficient.

The loss of hydraulic cylinder is mainly mechanical loss. The formula is:

$$P_{cf} = B_c \dot{X}_p^{\,2} \tag{18}$$

3. Parameter Influence Analysis and Parameter Determination Method

3.1 Parameter Influence Analysis

It is necessary to study the influence of load sensitive variable mechanism parameters on EHA energy loss and dynamics. Energy loss is calculated according to Chapter 2. The dynamics are characterized by the rise time of the step response. Firstly, the influence of the rate of displacement change on energy loss and dynamics of EHA was studied. Then the influence of parameters of load sensitive variable mechanism on the velocity of displacement change is studied.

The preliminary values of the load sensitive variable mechanism parameters are shown in Table 1:

Table 1 –Preliminary values of the parameters of the Load sensitive variable mechanism

Spring stiffness	Radius of damping hole	Piston area	Variable displacement starting pressure
20000N/m	$5x10^{-5}$m	$3x10^{-6}m^2$	14MPa

The following are the definitions of the two conditions

Condition A. The displacement is 40mm step command, and the load is the elastic force of the stiffness K_l =1800000N/m. $F_l = K_l X_p$.

Condition B. The displacement is a sine instruction with a amplitude of 28.8mm and a frequency of 0.4Hz, and the load is a sine force with a amplitude of 79000N and a frequency of 0.4Hz, one cycle. X_p = 28.8 sin (2 π x 0.4 t). F_l = 79000 sin (2 π x 0.4 t).

The influence of displacement change rate on EHA dynamics was studied under working Condition A. Take 80%, 90% and 100% of the displacement change rate respectively. As shown in the Fig. 3. The corresponding EHA rise time is shown in the Fig. 4 and Table 2. It can be seen that the larger the rate of displacement change, the longer the rise time of EHA, and the worse the EHA dynamic. This is because when the working condition is large load, the faster the displacement becomes smaller. The stiffness of the system decreases, resulting in decreased dynamic performance.

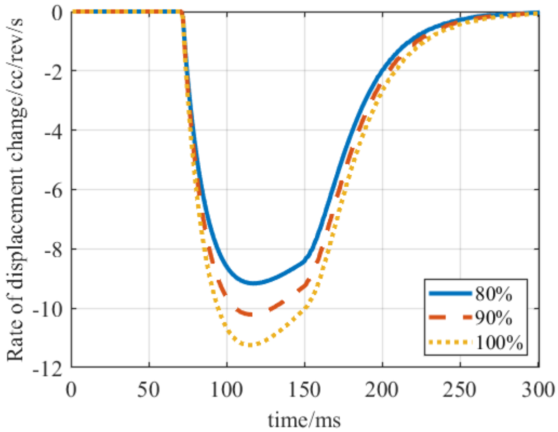

Figure 3 – Change speed of different displacement under working Condition A

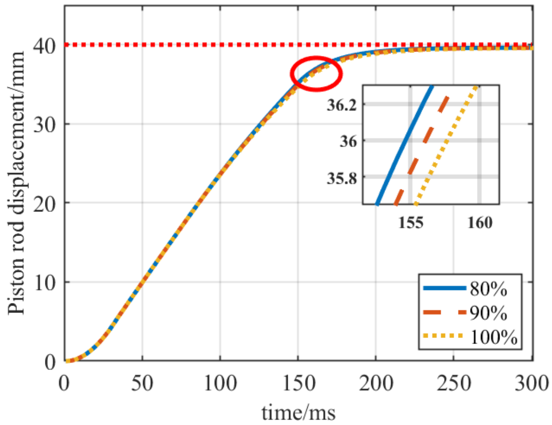

Figure 4 – Effect of displacement change rate on rise time under working Condition A

Table 2 –The risetime corresponding to the rate of displacement change

Rate of displacement change	80%	90%	100%
Risetime/s	0.154	0.156	0.157

The influence of displacement change rate on EHA energy loss was studied under working Condition B. Take 80%, 90% and 100% of the displacement change speed respectively, as shown in the Fig. 5. The corresponding EHA energy loss is shown in the Table 3. It can be seen that the larger the rate of displacement change, the smaller the EHA energy loss and the higher the EHA efficiency. This is because when the working condition is large load, the faster the displacement becomes smaller, the faster the motor copper loss is reduced, and the total energy loss is smaller.

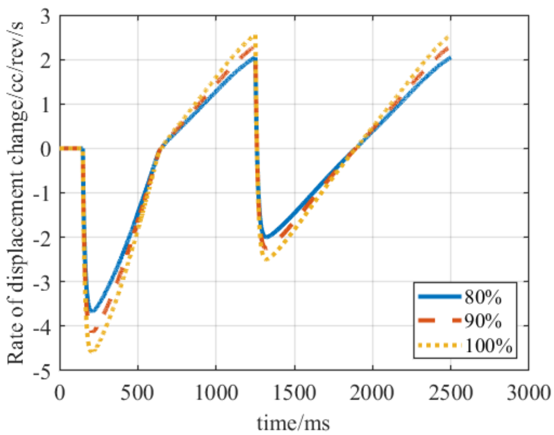

Figure 5 – Change speed of different displacement under working Condition B

Table 3 –Energy loss corresponding to the rate of displacement change under working Condition B

Rate of displacement change	80%	90%	100%
Energy loss/J	745	711	684

The effects of spring stiffness, damping hole radius, piston area and initial pressure on displacement velocity were studied under the working Condition A. The preliminary values of parameters of load sensitive variable mechanism are shown in Table 1. The impact results are shown in the Fig. 6-9. The final summary rule is shown in Table 4.

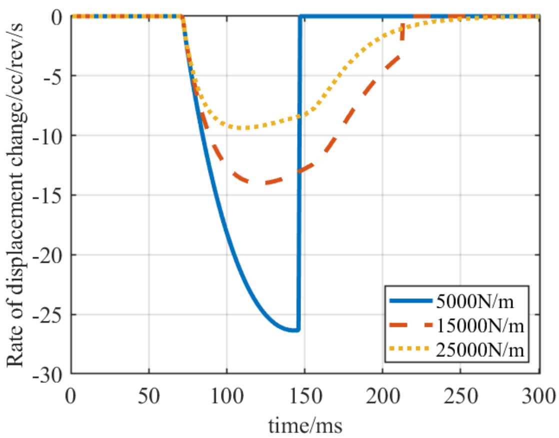

Figure 6 – Effect of spring stiffness on rate of displacement change

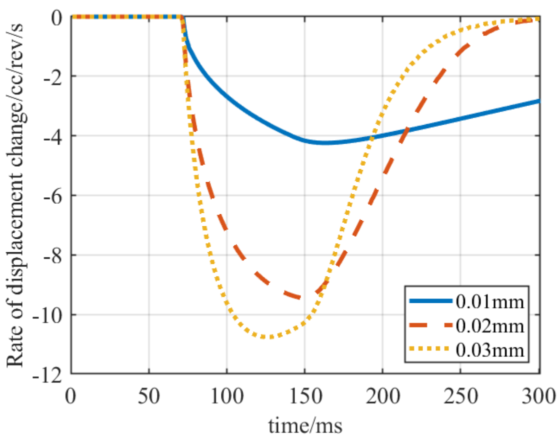

Figure 7 – Effect of radius of damping hole on rate of displacement change

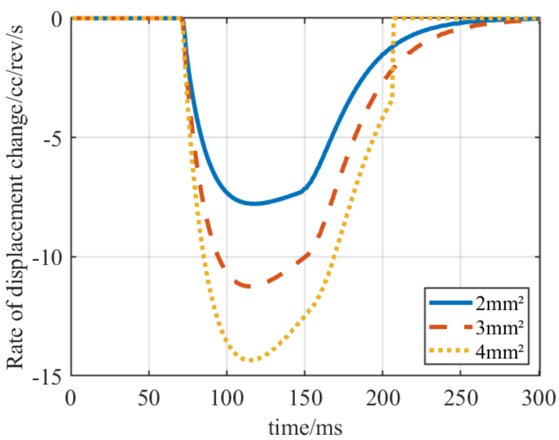

Figure 8 – Effect of piston area on rate of displacement change

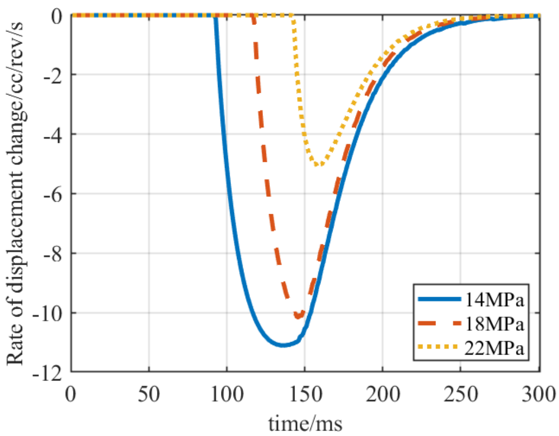

Figure 9 – Effect of initial variable pressure on rate of displacement change

Table 4 –Parameter influence result

	Rate of displacement change	Risetime	Energy loss
Rate of displacement change		proportional	inverse
Spring stiffness	inverse	inverse	proportional
Radius of damping hole	proportional	proportional	inverse
Piston area	proportional	proportional	inverse
Variable displacement starting pressure	inverse	inverse	proportional

3.2 Parameter Determination Method

It can be found that the four load-sensitive variable displacement mechanism parameters have opposite effects on EHA rise time and energy consumption. Therefore, the value of the parameter cannot be monotonically larger or smaller. Choose the right value in the interval so that the rise time and energy consumption are low. Because there are many variables, and the influence of different parameters on the rise time and energy consumption interact with each other. Therefore, particle swarm optimization algorithm can be used to determine the value of the four parameters.

Take risetime and energy consumption as design objectives. The expression is：

$$t_r = t_{90} - t_0$$
$$E_{EHA} = \int_0^{t_{\max}} \left(P_{Cu} + P_{Fe} + P_{mf} + P_{pf} + P_{pv} + P_{cf}\right) dt \tag{19}$$

Where t_r is risetime, t_{90} is the time when the piston rod displacement responds to the steady state value by 90%, t_0 is the time the piston rod is in the initial position, E_{EHA} is total energy consumption of the EHA.

Piston area, spring stiffness, radius of damping hole and initial pressure of variable displacement are taken as optimization variables. The number of particles is 40, and a total of 20 generations are optimized. The optimization results are shown in the Fig. 10.

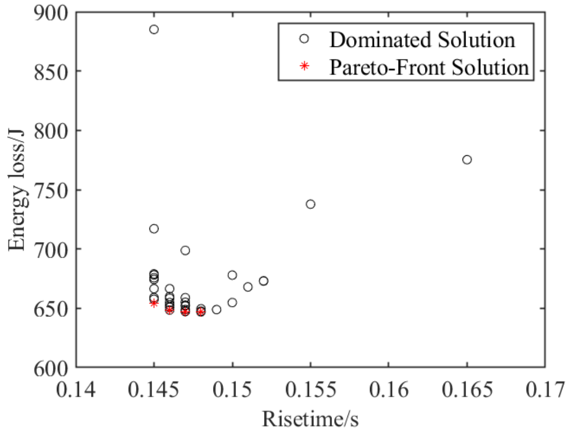

Figure 10 – Optimization result

Four dominant solutions were obtained after optimization. As shown in Table 3. Finally, domination solution 1 with more comprehensive effect is selected as the design reference value. The displacement response and displacement changes of dominant solution 1 under working Conditions A and B are shown in the Fig. 11-14. The EHA rise time without passive load sensing module is 0.145s. The displacement response is shown in the Fig. 15. The energy consumption is 1238J. It can be seen that passive load-sensitive EHA reduces the energy consumption of EHA under large load conditions on the premise of ensuring dynamic performance.

Table 5 – Dominant solution

Dominant solution number	Spring stiffness/ N/m	Radius of damping hole/m	Piston area/m^2	Variable displacement starting pressure/ MPa	Risetime/s	Energy loss/J
1	28366	10^{-4}	10^{-5}	24.4	0.147	647.09
2	30000	$1.3x10^{-4}$	10^{-5}	23.9	0.148	646.92
3	18011	10^{-4}	$9.5x10^{-6}$	27.3	0.145	654.44
4	23603	$1.2x10^{-4}$	$9.9x10^{-6}$	25.4	0.146	648.45

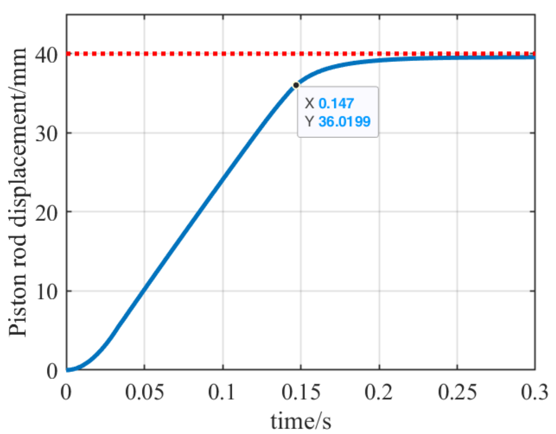

Figure 11 – Piston rod displacement response under Condition A

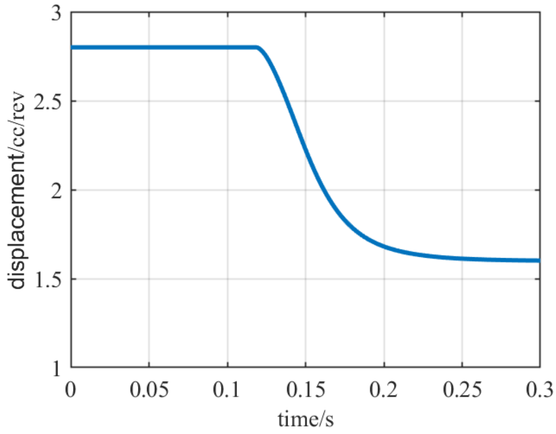

Figure 12 – Displacement change under working Condition A

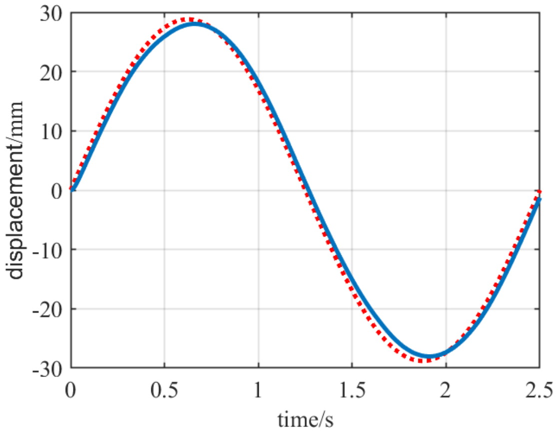

Figure 13 – Displacement response under Condition B

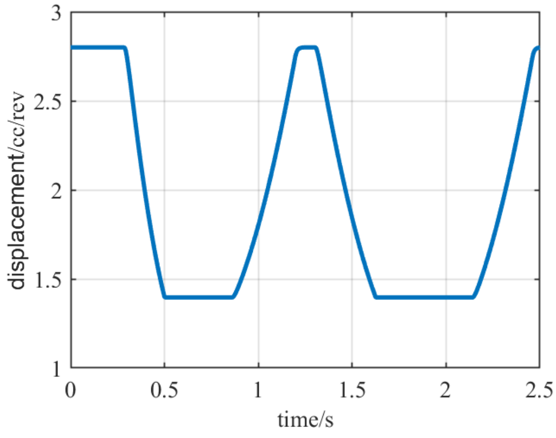

Figure 14 – Displacement change under working Condition B

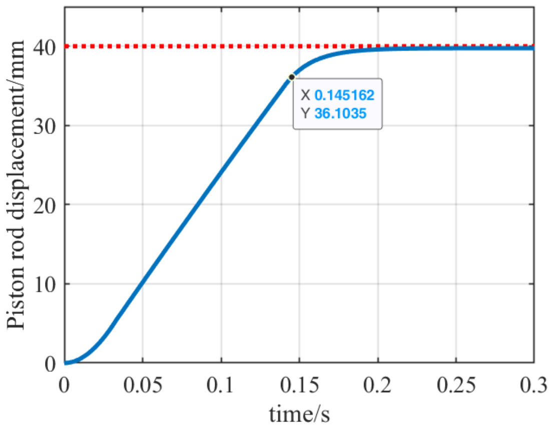

Figure 15 – No load-sensitive piston rod displacement response

4. Conclusion

The EHA energy consumption model is established. The effects of piston area, spring stiffness, radius of damping hole and variable initial pressure on EHA rise time and energy consumption of passive load-sensitive variable displacement mechanism were analyzed. The conclusion is that the spring stiffness and variable displacement initial pressure are inversely proportional to the rise time and proportional to the energy consumption. The radius of the damping hole and the area of the piston are proportional to the rise time and inversely proportional to the energy consumption. Particle swarm optimization is used to obtain a parameter set with smaller energy consumption and rise time. Under this parameter, the rise time is 0.147s and the energy consumption is 647.09J. Without the passive load sensing module, the EHA rise time is 0.145s and the energy consumption is 1238J. It can be seen that passive load-sensitive EHA reduces the energy consumption of EHA under large load conditions on the premise of ensuring dynamic performance. Future work can propose control algorithms for passive load-sensitive EHA. To solve the problem of dynamic decline in the process of emission reduction.

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6. Contact Author Email Address

Tian Yu : tony92565@sina.com

7. Copyright Statement

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