Design and Simulation of Electro-hydrostatic Actuator with a Built-in Power Regulator

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Abstract

The electro-hydrostatic actuator (EHA) is a kind of power-by-wire (PBW) actuator that converts the electrical power into localized hydraulic power for flight control. By removing the central hydraulic power supply together with hydraulic pipes, an EHA's reliability and efficiency are greatly improved but its frequency width and stiffness decreased. To overcome the drawback, this article proposes a novel structure of EHA associated with a power regulator. Composed of a high-pressure accumulator and a proportional valve, it can store and harness the hydraulic power flexibly according to the changing control requirements. The concept of transferred volume is put forward to estimate the capability of the power regulator. The actuator output position can be kept fixed with a hydraulic lock. The compounded control is specially developed to ensure the actuator system to operate in a correct manner. The simulation results indicate that the new-brand actuator results in efficient expanding of the system frequency width with an optimal power supply.

Keywords: electro-hydrostatic actuator (EHA); power regulator; transferred volume; hydraulic lock; frequency width; stiffness

1. Introduction

As an attractive technical concept for the next generation of aircraft, the power-by-wire (PBW) flight control system by removing central hydraulic power supply together with hydraulic pipes and using electrical cables for energy transmission, essentially increases the reliability and energy efficiency of an actuator system[1].

The electro-hydrostatic actuator (EHA) is one kind of PBW actuators that drives control surface of an airplane by localized hydraulic power transformed from the electrical power. According to the differences in control modes of the motors and pumps, EHA can be divided into three categories: namely, EHA with fixed pump displacements and variable motor speeds (FPVM-EHA), EHA with variable pump displacements and fixed motor speeds (VPFM-EHA) and EHA with variable pump displacements and variable motor speeds (VPVM-EHA). Of them, the FPVM-EHA is relatively popular thanks to its structural simplicity and high-efficiency[2].

In FPVM-EHA, a bi-directional pump rotates at variable speeds driven by an electrical motor. The reservoir is composed of a low-pressure accumulator and two check-valves to maintain the minimum pressure required by the system. The bypass-valve and relief-valve are arranged for system protection.

FPVM-EHA has found applications in a number of airliners, such as Airbus A380, but there are still some problems waiting for proper solutions. The maximum frequency width of a FPVM-EHA, which amounts only to 5 Hz, is not enough to meet the requirement for military uses. On the other hand, its stiffness is beneath what is expected[3-4] because the electrical motor continues to consume energy in the case of constant outputs thus making thermal dissipation rather difficult. This causes the needs for immediate solution to the problem concerning the effective use of the power of FPVM-EHA. A lot of researchers have attempted to solve it by developing high-performance electrical motors[5-7] and advanced control methods[8-10]. However, their effects are always limited by the property of electromagnetic materials[11].

This article presents a FPVM-EHA associated with a power regulator (FEPR), which improves the system performances through optimization of the supplied power regulator.
2. Structure Design

2.1. Configuration and operation modes

Fig.1 shows the schematic structure of a FPVM-EHA. Fig.2 shows the schematic structure of a FEPR. The power regulator in it is considered to be the second power source working in parallel with the FPVM-EHA. The proportional valve controls its output to provide transient flow. The hydraulic lock is a solenoid valve used to lock the actuator according to the operation modes and the system status. The bypass valve and the relief valve involved in FEPR are not shown in Fig.2.

There are three operational modes in FEPR, i.e.

1. High-efficiency mode, in which the power regulator does not work with the proportional valve at the median point and the system operates as a conventional FPVM-EHA itself.

2. High-speed mode, which, as a dual-control strategy to satisfy the expeditious requirements in short time, responds to the system more swiftly and the proportional valve this time controls the outputs of the power regulator to compensate the oil flow generated by the motor-pump.

3. Locking mode, in which the outputs of the actuator are kept unchanged by the solenoid valve and meanwhile, the motor-pump refeds the oil to the power regulator from the reservoir until the regulator’s pressure reaches the preset value.

2.2. Power regulator

The power regulator, which consists of a high-pressure accumulator and a proportional valve, stores and harneses the hydraulic power to ensure the high-maneuverability of flight.

The output flow of the power regulator helps accelerate or decelerate the actuator’s outputs. The additional flow enters the reservoir to balance the pressure difference between the power regulator and the reservoir. Fig.3 shows the transferring process, where the high-pressure and low-pressure accumulators should be designed carefully to guarantee the appropriate operating conditions. The bigger the total volume of accumulators, the more the power they can offer but the bigger and the heavier they would become.

The changes in pressure and volume can be calculated through the gas state equation and the flow continuity equation:

\[
P_h V_h^k = P_e (V_h + \Delta V_h)^k \\
\Delta V_h - \Delta V_i = \frac{V_i}{E_y} (P_e - P_i) \\
P_1 V_i^k = P_e (V_i - \Delta V_i)^k
\]

where \(\Delta V_h\) is the volume of output oil from the high-pressure accumulator, \(\Delta V_i\) the volume of input oil to the low-pressure accumulator, \(P_h\) the initial pressure of the high-pressure accumulator of the power regulator, \(V_h\) the initial gas volume of the high-pressure accumulator, \(P_i\) the initial pressure of the low-pressure accumulator of the reservoir, \(V_i\) the initial gas volume of the low-pressure accumulator of the reservoir, \(P_e\) the equilibrium pressure, \(E_y\) the total volume of the cylinder together with pipes, \(E_y\) the bulk modulus of elasticity of oil, and \(k\) the polytropic exponent of gas.

Compared to gas, the compressibility of oil is so small that it can be ignored, thus \(\Delta V_h \approx \Delta V_i\) and Eq.(1) can be simplified into

\[
P_h V_h^k = P_e (V_h + \Delta V)^k \\
P_1 V_i^k = P_e (V_i - \Delta V)^k
\]

where \(\Delta V\) is the transferred volume of the oil that can be obtained by

\[
\Delta V = \left(\sqrt[p_h - 1]{\frac{P_h}{P_i}} \right) \left(\frac{1}{1} / V_1 + 1 / V_i + 1 / V_h\right)
\]

As an important parameter to estimate the capability
of the power regulator, $\Delta V$ is a function of $p_h$, $p_1$, $V_h$ and $V_1$, where $p_h$ and $p_1$ are the maximum and minimum pressures of the system separately, so $\sqrt{p_h/p_1}$ can be considered to be a constant $N$ ($N > 1$).

Rewrite Eq.(3) into

$$\Delta V = \frac{N - 1}{(N - 1)/V_1 + 1/V_1 + 1/V_h}$$

(4)

Eq.(4) indicates that it will be more effective to increase $V_1$ rather than $V_h$ to augment $\Delta V$.

2.3. Hydraulic lock

The hydraulic lock works in cooperation with the power regulator of the system to effectuate the two following functions: ① If the controller discovers that the outputs are kept constant and the static error is less than the specified value, the hydraulic lock isolates the cylinder from the system. Therefore, the external load can no longer act on the electrical motor. ② The motor refeeds the power regulator from the reservoir, if necessary.

Once a change takes place in the external load or position input, the actuator gets unlocked immediately.

3. System Modeling

Since there are many nonlinear factors in the system, a transfer function is not sufficient to describe their performances. This article uses a block diagram modeling method on the basis of the equations of a brushless direct current (DC) motor as follows[12]:

$$u_m = E + L \frac{di}{dt} + R i$$

$$E = K_c \omega$$

$$i = \frac{T_e}{K_t}$$

$$T_e = J \omega + K_vm \omega + T_l$$

where $u_m$ is the input voltage of motor, $E$ the back electromotive force, $L$ the inductance of motor, $R$ the resistance of motor, $i$ the current of motor, $K_c$ the speed constant of motor, $\omega$ the rotational speed, $T_e$ the electromagnetic torque, $K_t$ the torque constant of motor, $J$ the inertia of motor, $K_vm$ the viscous coefficient of motor, and $T_l$ the load torque.

Fig.4 shows the block diagram model of the motor according to Eq.(5), where $s$ is the Laplace operator. The part framed by dash lines represents a current protective device made of software.

The in/out flows of the pump are

$$Q_{in} = D \omega - K_{ilp}(p_1 - p_2) + K_{elp}(p_2 - p_1)$$

(6)

$$Q_{out} = D \omega - K_{ilp}(p_1 - p_2) - K_{elp}(p_1 - p_2)$$

(7)

where $D$ is the displacement of pump, $p_1$ the inlet pressure of pump, $p_2$ the outlet pressure of pump, $p_r$ the pressure of reservoir, $K_{ilp}$ the internal leakage coefficient of pump, and $K_{elp}$ the external leakage coefficient of pump. Fig.5 illustrates the diagram model of a pump, where $Q_{el}$ is the external leakage of pump.

In order to ensure the flow balance between the cylinder and the pump, the EHA should be of symmetrical type. The chamber of a cylinder is divided by the piston into two working volumes (see Fig.6).

The flow in Volume 1 can be described by

$$Q_1 = A \dot{X}_p + \frac{V_{10} + AX_p}{E_y} \dot{p}_1 + K_{il}(p_1 - p_2)$$

(8)

The flow in Volume 2 can be described by

$$Q_2 = A \dot{X}_p - \frac{V_{20} - AX_p}{E_y} \dot{p}_2 + K_{il}(p_1 - p_2)$$

(9)

where $A$ is the active area of piston, $K_{il}$ the internal leakage coefficient of cylinder, $X_p$ the position of the piston, $V_{10}$ and $V_{20}$ are the initial volumes of both Volume 1 and Volume 2, which are equal in value due to the actuator’s nature of symmetry.
The load force equation of the piston is

\[ A(p_1 - p_2) = M\dot{X}_p + K_{vp}\dot{X}_p + F_{ex} \quad (10) \]

where \( M \) is the total mass of piston and load, \( K_{vp} \) the viscous coefficient of piston and \( F_{ex} \) the external load force.

Fig.7 and Fig.8 diagrammatize the models of the cylinder and the piston respectively.

4. Control Design

The controller receives a position command and the sensor feedbacks it to generate the control signals. The inputs of the electrical motor and proportional valve are continuous signals while the input of the hydraulic lock is an on-off signal (see Fig.10).

Fig.11 illustrates the principles of mode switching, where \( r \) is the position command. If the static error \( r - X_p \), velocity \( \dot{X}_p \) and acceleration \( \ddot{X}_p \) are below the preset thresholds, the actuator is locked. The moment they get over, the system comes into operation. The shift from the Mode A to the Mode B depends on the value of command derivative \( \dot{r} \).

4.1. Motor control

In the Mode A, the electrical motor serves as the only power source in the system while in the Mode B, the electrical motor should work in cooperation with the power regulator.

The state feedback matrix places the closed-loop poles of the system at desirable locations in the complex plane of Laplace transforms. These poles correspond to the system eigenvalues, which determine the characteristics of response.

Consider the FEPR as a fifth-order system with following state variables: \[ T \]

\[ X = \begin{bmatrix} i & \omega & p_t & X_p & \dot{X}_p \end{bmatrix}^T \quad (12) \]

where \( p_t \) is the pressure difference between the two volumes of the cylinder. According to Eqs.(5)-(10), the state equations of EHA can be described in a matrix form:
$K = \begin{bmatrix} R & -K_e & 0 & 0 \\ L & -K_m & 0 & 0 \\ J & K & 0 & 0 \\ 0 & 2E_1D & -2E_1K & 0 -2AE_1 \\ 0 & V_10 & 0 & 0 \\ 0 & 0 & A & 0 \\ 0 & 0 & 0 & K_{vp} \\ 0 & 0 & 0 & M \\ \end{bmatrix} \begin{bmatrix} 1 \\ L \\ -B \\ u_m \end{bmatrix}$ \hspace{1cm} (13)

Output vector: $Y = \begin{bmatrix} 0 & 0 & 0 & 1 \end{bmatrix} X$ \hspace{1cm} (14)

where $K_t = 0.5K_{lep} + K_{lp} + K_{hp}$ is the total leakage coefficient of the pump and cylinder.

It is convenient to calculate the state feedback matrix with MATLAB. In order to eliminate the static error caused by the pressure feedback, a high-pass filter is applied to the pressure feedback loop.

In the Mode C, to refeed the power regulator, the power regulator is unworkable (Mode A).

4.2. Proportional valve control

The differential of position command $r\dot{e}$ indicates the requirement for speed used to determine which Mode the system works in: A or B.

The proportional valve is controlled by the proportion differential (PD) method as follows:

$$u_p = G(\dot{e})(K_p\dot{e} + K_d \frac{de}{dt})$$ \hspace{1cm} (15)

where $u_p$ is the input of proportional valve, $e$ the position error of system, $K_p$ the proportional coefficient, $K_d$ the differential coefficient and $G(\dot{e})$ the variable gain, which is

$$G(\dot{e}) = \begin{cases} 0 & |\dot{e}| < b \\ K |\dot{e}| & |\dot{e}| \geq b \end{cases} \hspace{1cm} (16)$$

where $b$ is the switching velocity and $K$ the gain of $|\dot{e}|$. If $|\dot{e}|$ is greater than $b$, the power regulator is workable and controlled by $u_p$ (Mode B), or else, when $u_p=0$, the power regulator is unworkable (Mode A).

In the Mode C, the proportional valve opens to maximum to fuel the power regulator and the power regulator, pump and reservoir are connected with each other. If the pressure of power regulator reaches $p_{es}$, the proportional valve is closed.

4.3. Hydraulic lock control

The actuator can be locked once its output has attained the desired value. The strategy to make decision for this is:

(1) If position error $e$ is lower than the specified value $E_1$, i.e., $e < E_1$; 
(2) If the velocity of the piston $\dot{X}_p$ is lower than the specified value $V_1$, i.e., $\dot{X}_p < V_1$ and
(3) If the acceleration of the piston $\ddot{X}_p$ is lower than the specified value $A_1$, i.e., $\ddot{X}_p < A_1$.

Then, the controlled input is

$$u_n = \begin{cases} 0 & e < E_1, \dot{X}_p < V_1 \text{ and } \ddot{X}_p < A_1 \\ 1 & \text{else} \end{cases} \hspace{1cm} (17)$$

Usually, $E_1$, $V_1$ and $A_1$ are close to zero, which, however, should not be overly small to avoid the fault-switching. Therefore, it is urgent to acquire the appropriate thresholds through simulation and experiments.

For the purpose to deduct the $\dot{X}_p$ and the $\ddot{X}_p$ from the $X_{vp}$, is used the nonlinear tracking-differentiator[14-15], which allows to obtain the results more accurate than other methods.

5. Simulation and Analysis

Table 1 lists the system parameters of FEPR. Fig.12 compares the step responses of the FEPR and the conventional FPVM-EHA. The external force of 30 000 N (30% maximum) is acted on the system for 2.5 s to observe the stiffness.

![Fig.12 Position responses.](image)
To find out the proper size of the two accumulators in Fig.3, a group of comparison tests have been carried out with different $V_h$ and $V_l$ in Table 2.

The transferred volumes for each testing calculated with Eq.(4) are listed in Table 3 and Fig.14 shows the corresponding simulation results.

**Table 2 Parameters for testings**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_h$/mL</td>
<td>300</td>
<td>600</td>
<td>300</td>
</tr>
<tr>
<td>$V_l$/mL</td>
<td>300</td>
<td>300</td>
<td>600</td>
</tr>
<tr>
<td>$N$</td>
<td>7.61 ($p_h = 28$ MPa, $p_l = 2$ MPa)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Table 3 Transferred volumes**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta V$/mL</td>
<td>230.3</td>
<td>244.5</td>
<td>412.8</td>
</tr>
</tbody>
</table>

It is clear that the efficient way to obtain the bigger transferred volume is to increase $V_l$ rather than $V_h$, which means the power regulator can make more contribution towards the system.

Fig.15 exhibits the tracking performances of a FEPR. It discovers that there is little difference between Test 1 and Test 2 and the output can track the input for quite a long time in Test 3 because of the bigger transferred volumes.

**6. Conclusions**

The proposed FEPR is a new commitment to the high-performance actuator system, which meets a variety of control requirements. In the high-efficiency mode, it operates the same as the FPVM-EHA; in the high-speed mode, it uses the associated power regulator to gain fast responses and expand frequency width. Moreover, the hydraulic lock helps to preserve the stiffness and reduce the consumed energy when loaded. Both calculated and simulation results indicate the optimization principle of transferred volume for power regulator.

In the future, it is expected to fulfill the global optimization in terms of performances, size, weight and thermal dissipation by regarding the motor, the pump and the power regulator as an integral. The hydraulic lock can be integrated into the bypass-valve for size reduction. The high robust control method along with some intelligent strategies is now under development.
References


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基于能量调节的电动静液作动器设计与仿真
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摘 要: 电动静液作动器（EHA）是一种功率电传（PBW）作动器，可将电能输入转换为局部液压能以实现飞行控制。通过取消中央液压源和液压管路，电动静液作动器的可靠性和效率获得提高，但是带宽和刚度受到限制。为解决上述问题，本文提出了一种基于能量调节的新型电动静液作动器。能量调节器由高压蓄能器和比例阀组成，可以根据不同的控制需求储存和管理液压能。提出了转移体积的概念用于评价调节器的工作能力。液压锁用于保持作动器的输出位置。仿真结果表明：经过优化的新结构有效扩展了系统频宽。

关键词: 电动静液作动器; 能量调节器; 转移体积; 液压锁; 频宽; 刚度