Study of a throttle work-fluid flow regulations characteristics of the electro hydraulic actuator under aerodynamic load

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Abstract. This article addresses the issue of external load influence on characteristics of electrohydraulic actuator throttle regulation. Hinge moment, constant and inertial load as well as viscous friction and working fluid compressibility were considered as an external load. In research of «ultimate» dynamics actuator’s capabilities external load has the greatest impact, thus actuator’s bandwidth is reduced and phase delay is increased. The result of these influences on static and dynamic characteristics is proposed to be called «the throttle effect» in electro hydraulic actuators. Taking into account «throttle effect» of electrohydraulic actuator’s mathematical model was obtained.

1. Introduction
First aircraft control systems were embodied in the form of mechanical connection between controls and surfaces. External air force impact on control surface was on the increase with the gradual growth of flight speed that led to appearance of aerodynamic trimmers allowing to reduce external load of airflow for pilot work facilitation. However, aerodynamic trimmers was not enough for control, due to transition on supersonic flight mode and first hydro mechanical actuators– boosters, therefore, appeared in pilot-surface loop[1]. Apart from the fact that flight-control systems actuators perform the usual function of actuators, moreover, they are essentially single mechanical links between an object and control surface.

Historically, actuators can be classified into the following categories [2]:
1. By the type of actuating medium using for motion creation:
   - electric,
   - hydraulic,
   - pneumatic.
2. By the nature of output link motion:
   - progressive motion,
   - rotary motion.
3. By the type of input signal:
   - analog control,
   - discrete control,
   - relay control.

We are interested in the category according to the type of actuating medium, since in fact of what actuating medium type will be used, such will be the architecture of actuator. At the moment, there are concepts of creation «more electric aircraft» or «fully electric aircraft». In the article [3] there were
considered main disadvantages while using electromechanical actuators in primary control system and possible methods to eliminate these lacks.

Today in primary control system of maneuverable aircrafts there are used electro hydraulic actuators which have the main advantages of electric control and hydraulic power unit in their concept. There exist three architecture in design of electro hydraulic actuation systems: «summation of force», «summation of rates» and «summation of movement». [4] In modern maneuverable aircrafts electro hydraulic actuator systems designed by «summation of force» architecture are established in primary control systems [5, 6]. The article [7] contains information of force-fight in parallel redundant actuation systems developed by «summation of force» architecture without research of external load impact and actuator’s internal loop dynamics. The designing of electro hydraulic actuators must be considered the external load influence on actuators dynamic properties [8]. As for the creation of control systems, that identifies actuators systems channels failures, it’s difficult to create at the stage of system design [9]. There were written a lot of scientific studies about the creation impediments of complex system of control [10-13]. For the of right system of control «the fault tree» should be created on the part of designing and testing [10], i.e. from the point of view of reliability the most «weakest» elements of the system should be shown on this scheme.

Today slightly stable and unstable aerodynamic layouts are widely used in modern maneuverable aircraft design, thus traditional mechanical wiring has been replaced by Fly-by-wire flight controls systems. Hydro mechanical actuators (boosters) were changed by electro hydraulic throttle regulation actuators. At equal power on the steering aircraft’s surface, electro hydraulic throttle regulation actuators have the smallest overall mass characteristics.

Electro hydraulic actuators should be considered not only as a constructive part of the object: «actuators – control surface», but also as fundamental part of any control system, inasmuch as on communication schemes base of actuators with object, their interaction schemes and their «ultimate» dynamic capabilities, all of this is a base for architect designing of each Fly-by-wire flight controls systems[14].

2. Problem definition

The transition to slightly stable and unstable aerodynamic layouts of aircrafts leads to the creation of Fly-by-wire systems and electro hydraulic actuator throttle regulation (EHA), which provides minimization overall mass characteristics at the required power. Electro hydraulic actuator’s functional model is shown on the figure 1 [15]. In fly default external load has an impact on EHA, which determines dynamic capabilities of actuator. In this paper that effect has been studied.

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**Figure 1.** Functional model of electro hydraulic throttle regulation actuator.
U_in – input control signal, k_a – fault amplifier gain, SOV – solenoid valve, EHSV – electro hydraulic servovalve, x_{sv} – slide valve (spool) displacement with respect to housing, k_{LVDT} - linear variable differential transformer gain, LVDT – linear variable differential transformer, x_p – piston displacement with respect to cylinder, r – lever arm of aerodynamic force; M_{hn} – moment of control surface, which are proportional to the angle \phi of control surface, F_p – load force on piston, Qin – fluid flow into hydro cylinder chambers; p_in – input pressure; p_out – output pressure, p_1 – hydro cylinder first chamber pressure, p_2 – hydro cylinder second chamber pressure.

Usually, linear dynamic actuator model is used for Fly-by-wire control system algorithm design without taking into consideration external load. That type of linear model is shown on the figure 2.

\[
\begin{align*}
U_{in}(p) & \rightarrow U_0(p) \rightarrow k_a k_{EHSV} \frac{T_{EHSV} p + 1}{p} x_{sv}(p) \rightarrow k_{LVDT} r \\
& \downarrow \quad U_{LVDT}(p) \downarrow \quad k_{LVDT} r' \quad \varphi(p)
\end{align*}
\]

**Figure 2.** Linear dynamic actuator model without taking into consideration external load.

\[ T_{EHSV} \text{ - electro hydraulic servovalve time constant, } k_a, k_{EHSV}, k_{LVDT} \text{ - electrical amplifier gain, electro hydraulic servovalve gain, no load slide valve (spool) fluid flow gain, linear variable differential transformer gain.} \]

Linear model coefficient product represent actuator’s rate dynamic:

\[ D = k_a k_{EHSV} k_{LVDT}, \]

and define actuator’s dynamic characteristics, bandwidth, performance, overshoot.

Under external load on steering surface, that is represented by the summation of inertial moments, hinge moment and viscous flow friction, there is used linear model that is shown on the figure 3, when non-linear equation of flow through control throttle is expandable in the Taylor’s series and the first terms of series are taken into account. It is also considered that dynamic process is going in the vicinity of a stationary actuator motion at middle piston position into hydro cylinder.

\[ \Delta Q_e = (dQ/dx_{sv}) \Delta x_{sv} + (dQ/dp_{p}) \Delta p_{p}, \]

where \( Q_e \) - hydro cylinder port valve fluid flow, \( p_{p} \) - load pressure, \( x_{sv} \) - slide valve (spool) displacement with respect to housing

\[
\begin{align*}
A_p & \rightarrow \dot{\lambda}_p \rightarrow \lambda_p \rightarrow T_{EHSV} r + 1 \\
& \downarrow \quad A_p \rightarrow K r \quad 1 \quad \phi \quad 1 \quad s \quad \varphi \\
& \downarrow \quad 2E \quad 1 \quad \phi \quad 1 \quad s \quad M_{hn} \\
& \downarrow \quad M_{hn} \rightarrow k_{if} \rightarrow C_{hn} \\
& \downarrow \quad k_{LVDT} x_p \rightarrow r \\
& \downarrow \quad k_{LVDT} r' \rightarrow \varphi(p)
\end{align*}
\]

**Figure 3.** Linear model with regard to viscous friction, moment of inertia and hinge load.
In this paper, non-linear dependency of working fluid flow through hydraulic slide valve is used for design of non-linear mathematical actuator’s model:

\[ Q_{sv} = G(x_{sv}) \sqrt{p_{in} - p_f \text{sign} x_{sv}}, \]  

(3)

where \( Q_{sv} \) - slide valve (spool) fluid flow, \( G(x_{sv}) \) - admittance dependency of slide valve (spool) displacement with respect to housing, \( p_{in} \) - input pressure.

Studding non-linear dependence of working fluid flow on pressure drop, it is essential to be considered the external load influence is studied on the characteristics of electro hydraulic actuator throttle regulation in whole relation of slide valve displacement to load pressure and load flow region. In this case we shall write the equation of fluid flow through hydraulic slide valve \( Q_{sv} \) and the equation of fluid flow into hydro cylinder chamber \( Q_{fc} \) taking into consideration continuous of flow (\( Q_{sv} = Q_{fc} \)):

\[ \frac{2}{\mu} \sqrt{p_{in} - p_f \text{sign} x_{sv}} = G(x_{sv}) \sqrt{p_{in} - p_f \text{sign} x_{sv}} = Q_{fc}, \]  

(4)

where \( \mu \) - rate of flow coefficient, \( A_0(x_{sv}) \) – throttling orifice area, which depends on slide valve (spool) displacement with respect to housing and orifice geometry form, \( G(x_{sv}) \) – admittance dependency of slide valve (spool) displacement with respect to housing.

With the alteration of slide valve (spool) position, throttling orifice area is changed too, that leads to reversal of admittance dependency of slide valve (spool) displacement. Thus, fluid flow supplies hydro cylinder chamber \( Q_{fc} \), transmits the rate to actuator’s piston \( Q_{rp} \), compensates the \( Q_{lf} \) and compensates the compressibility of liquids \( Q_c \):

\[ Q_{fc} = Q_{rp} + Q_{lf} + Q_c = A_r \frac{d\phi(t)}{dt} + q_f p_f + \frac{A H dp_f}{2E dt}, \]  

(5)

where \( A_r \) - ram area, \( r \) - lever arm radius, \( \phi \) - control surface deflection angle, \( q_f \) - leakage of flow gain, \( H \) - full travel of piston, \( E \) - bulk modulus of elasticity of the fluid.

Provided continuously of fluid flow: fluid flow of slide valve (spool) fully come in hydro cylinder chambers.

Then, the equation (5) may be written in operator’s form:

\[ Q_{fc} = A_r r \phi(t) + q_f p_f (1 + T_h s), \]  

(6)

where \( T_h \) - hydraulic time constant.

The equation of moments (load) has the following expression: moment of actuator \( M_a \) overcomes hinge moment \( M_{hm} \) of control surface, inertial moment \( M_i \) and friction moment \( M_f \) :

\[ M_i + M_f + M_{hm}^2 = M_a \text{ or } I_s \ddot{\phi} + k_{sf} \dot{\phi} + M_{hm}^2 \phi_{sf} = A_r r p_f, \]  

(7)

where \( M_i \) - inertial moment of control surface, \( k_{sf} \) - viscous friction gain

Solving equations (4), (6) and (7) together and translating differential equation into operator form \( d/dt = s = j\omega \), we get the following equation:
where $x_p$ - piston displacement with respect to cylinder, $V = A_t H$ - volume of «working» hydro cylinder’s chamber, $m$ - mass of control surface, $C_{hm}$ - hinge moment gain.

The above expression led to study process that occur in electrohydraulic actuator throttle regulation when different types of load is exposed.

3. The research and results

As a first step, the influence of constant load $F_p = A_r P_r = \text{const}$ on actuator’s piston is being study.

On the figure 4 (a) the load is represented by the straight line shifted relatively to y-axis by $p_f$. As a result actuator’s speed ability has significantly changed, it has become asymmetrical relatively y-axis (fig. 4a).

Counteracting load reduces dynamic actuator capabilities, assisted increases (fig. 4b).

Studying frequency response under harmonic and positional load, one half-wave of sine wave has a phase delay and the ratio of output value ($U(\psi)$) to input signal ($U_r$), another half-wave has completely different phase delay and ratio (fig. 5 and 6). On figure 5, amplitude of impute signal ($U_r$) is $A = 25^\circ$, frequency $f = 1$ Hz, positional load is 75% of maximal load. Amplitude of output link deflection ($\psi$) does not correspond to harmonic input signal. Output signal of piston motion deforms due to the load influence ($\Delta \psi_{\text{ass}}$ - input and output amplitude difference (odds) under assisted load, $\Delta \psi_{\text{cl}}$ - input and output amplitude difference (odds) under counteracting load). The effect of assisted load is (fig. 6) reducing to phase delay (curve b) regarding to actuator without load (curve a), if load is counteract phase delay is increasing (curve c). That deformation of characteristics we will call «throttle effect» in electro hydraulic actuation systems.

Figure 4. The change of rate characteristic under constant positional load
If the deflection rule of control surface and accordingly external load is harmonic, equation of required speed and moment for this operating mode will take the form of ellipse equation that is rotated by y-axis due to a friction.

\[
\begin{align*}
\dot{\phi}_{cs}(t) &= A\omega \cos \omega t \\
M_l(t) &= (M_{hm}^e - I\omega^2)\dot{\phi}_{cs}(t) = A\sin \omega t + k_{\phi} \omega A \cos \omega t \\
M_l^2 &= (M_{hm}^e - I\omega^2)A^2(1 - (\dot{\phi}_{cs}/\omega)^2) + k_{\phi}^2 \phi_{cs}^2,
\end{align*}
\]

(9)

(10)

In this case, the dependences become more complex. Rate plots have loop-shaped form, depending not only on the type and the magnitude of the load, but also on shape, amplitude and frequency of input signal (fig.7) as well as from rate plots is depending on initial condition, under which dynamical processes are studied.

The study of non-linear model of electro hydraulic confirms the equation above. On the figure 8 (a) the «ellipse of load» is shown in coordinate of slide valve (spool) displacement to load pressure and load flow for input signal amplitude ±5 mm and ±15 mm and frequency 0.5 Hz. On the figure 8 (b) – corresponding change of rate characteristics.

![Figure 5](image1.png)  
**Figure 5.** Output link displacement with respect to cylinder distortion in consequence of «throttle effect».

![Figure 6](image2.png)  
**Figure 6.** The comparison of phase delay characteristics of linear model with respect to «throttle effect» under different load.

![Figure 7](image3.png)  
**Figure 7.** The alteration of actuator’s rate characteristic with respect to friction and compressibility of liquids.
On the figure 9 (a) the «ellipse of load» is shown in coordinate of slide valve (spool) displacement to load pressure and load flow for input signal amplitude ±15 mm and frequency 0.5 and 1 Hz. On the figure 9 (b) – corresponding change of rate characteristics. As can be seen the «ellipse of load» rotated with increasing of frequency due to the inertial load, which has a quadratic dependence on frequency of the input signal according to equation 11.

4. Conclusion
Mathematical model of electro hydraulic actuator throttle regulation study represents the influence of different types of load on static and dynamic actuator’s characteristics. The result of this influence is the deformation of actuator’s characteristics that is proposed to be called «the throttle effect» in electro hydraulic actuator.

«The throttle effect» presence translates electro hydraulic actuators throttle regulation at the systems category with variable parameters which are depending on type, amplitude and frequencies of input signals determining the state of aircraft control system and load mode as also as the initial condition of beginning of dynamic process.

It was found that «the throttle effect» has the greatest impact when counteracting load value exceeds 50% of maximal actuator’s force. In this case, the rate of piston drops over 30%.
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