Journal of Physics: Conference Series

PAPER • **OPEN ACCESS**

Simulation of an Aerospace Electrohydraulic Servomechanism, with different Coulomb Friction models

To cite this article: P Alimhillaj *et al* 2024 *J. Phys.: Conf. Ser.* **2716** 012052

Simulation of an Aerospace Electrohydraulic Servomechanism, with different Coulomb Friction models

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Abstract: The contemporary design of flight control systems demands the utilization of intricate models for the in-depth analysis of individual components or subsystems. Conversely, there is a parallel need for more foundational and synthetic models that offer sufficient accuracy, specifically tailored for preliminary design, monitoring, or diagnostic purposes.
This paper centers on electro-hydraulic servomechanisms designed for aeronautical applications, emphasizing their contemporary significance. These systems play a crucial role, particularly in primary flight commands characterized by precise position servo control.
The great variety of configurations and applications, their complexity and the criticality that characterizes this servomechanisms, deemed appropriate to devote particular attention to their modeling and the development of numerical simulation systems models that are versatile and reliable (flexible and easily applicable to different real systems but capable of providing realistic simulations). In particular, in this work are presented two innovative Coulomb friction models which are applied through MATLAB/Simulink block diagram structure to the model of the electrohydraulic servomechanism. The two friction models are foreseen to overcome the problematic of standard models for the friction, giving more realistic results, increasing the accuracy of the simulations.

1. Introduction
The design of flight control systems necessitates highly detailed models. One of key problems is to develop a valid and versatile formulation of the Coulomb friction phenomena acting on an entire servomechanism or in its subsystems, and to acquire in advance the appropriate numerical simulation models of its various components which contribute to the realization of the aforementioned system.
The servomechanism to set up the friction models and the simplified numerical models of the complete system, is a typical pure powered electro-hydraulic position servo, which finds wide application in flight controls whether they are of primary or secondary type [4].
The servomechanism analysed is composed of:
1 - PID position control logic (proportional - integrative - derivative) modifiable in PI or PD or proportional with speed loop
2 - two-stage electro-hydraulic servo valve or “flapper nozzle”
- Linear actuator

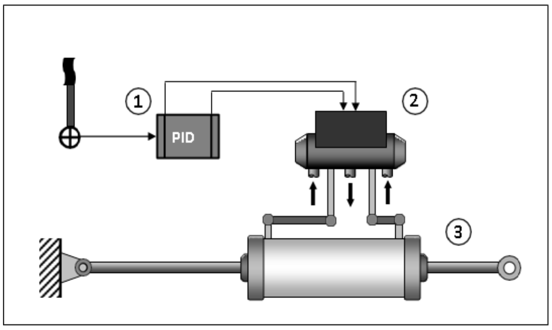

Figure 1. Schematic representation of the considered hydraulic system.

2. Mathematical model

We can use Matlab - Simulink as methodology to describe dynamic systems and to examine their peculiarities and response characteristics of a typical pure enhanced electro-hydraulic position servomechanism system, that nowadays find a very widespread use in flight controls and on-board mechanisms [5-6].

2.1 PID Control logic

The signals produced by the system, added algebraically to those coming from the feedback loop (which gives the instantaneously position assumed by the servomechanism and to compared it with the commanded one), make possible to define an instantaneous position error signal of the commanded surface. The position error obtained in this way is transformed by the position control logic, for example PID, into the driving current which, by operating on the electro-hydraulic servo valve, generates the actuation of the command originally given by the pilot and by continuously pursuing the reduction of the absolute value of the position error [1-14].

2.2 Two-stage electro-hydraulic servo valve or "flapper nozzle"

2.2.1 *Electromechanical model.* It is assumed that the driving current "command" Cor is immediately translated, into a force which, interacting with the centering action produced by the spring KFS, gives the spool move XS. Given that the valve spools used in aeronautical applications often have extremely small dimensions and that the damping and inertial actions to which they are subject are equally modest, as a first approximation, can be neglected, reducing the system with an instantaneous dynamics. Therefore the equation of the spool is:

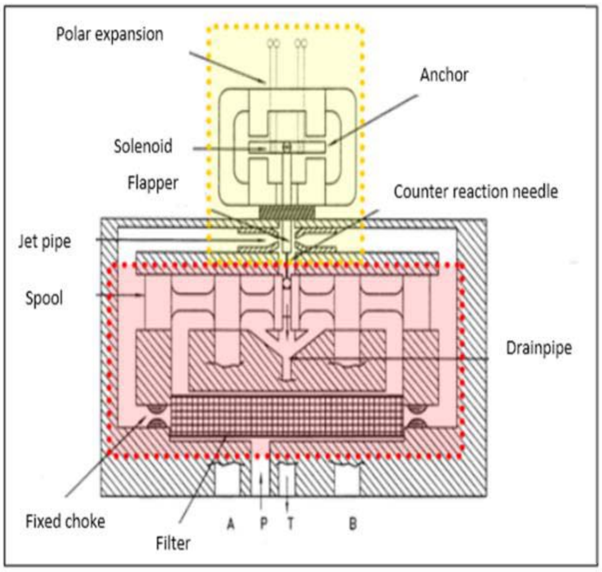

Figure 2. Schematic representation of a flapper - nozzle type servo valve

$$F = Cor \cdot GM = XS \cdot KFS \quad \Rightarrow \quad XS = \frac{GM}{KFS} \cdot Cor \tag{1}$$

2.2.2 *Fluid dynamics model of the servo valve.* For small valve spool openings, the curves in Fig. 4 can be approximated with straight lines of minimum deviation passing through the origin and having angular coefficients GP and GQ which can be read directly on the graphs; it can define linear relationships capable of modeling, the fluid dynamics behavior with sufficient accuracy.

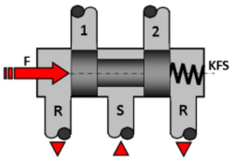

Figure 3. Electromechanical model of the servo valve

2.3 Linear Actuator

Figure 5 shows a double-acting symmetrical piston with no leakage through the sealing gaskets. A dynamic model characterized by inertia, viscous friction, aerodynamic load, and driving force was used in the actuator's numerical simulation program.

The dynamic equilibrium equation of the actuator is the follows:

$$MJ \cdot \frac{d^2 XJ}{dt^2} + CJ \cdot \frac{dXJ}{dt} = AJ \cdot P12 - FR \tag{2}$$

Where it is possible to obtain the F.d.T. of the actuator as:

$$\frac{d^2 XJ}{dt^2} = \left(AJ \cdot P12 - FR - CJ \cdot \frac{dXJ}{dt} \right) \Big/ MJ \tag{3}$$

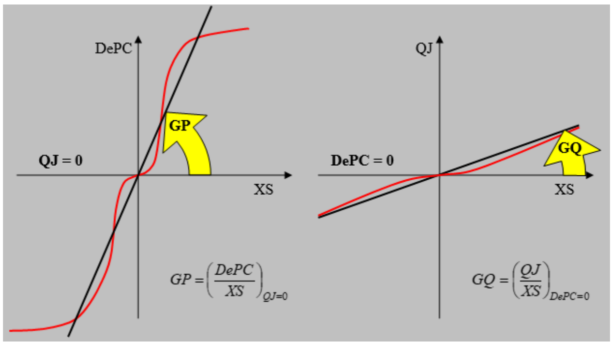

Figure 4. Typical trend (in red) of the DePC – XS and QJ – XS curves with the relative ones linearized curves (straight lines in black) and the linearized coefficients GP and GQ

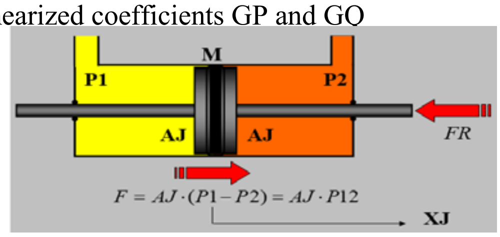

Figure 5. Schematic representation of the linear actuator with external loads FR

2.4. Coulomb friction calculation routine

The Coulomb friction model used, introduces a contribution into the rod that, with a good approximation, simulates the real operating conditions of the actuator, with starts, stops, steps and positioning errors. In order to compare the proposed Coulomb friction model and the related calculation algorithm with those commonly used in the most commercial programs suitable for dynamic simulation. Some cases relating to particularly significant functional conditions are presented here after in different variants obtained by applying the different models considered to the same dynamic system [8-10].

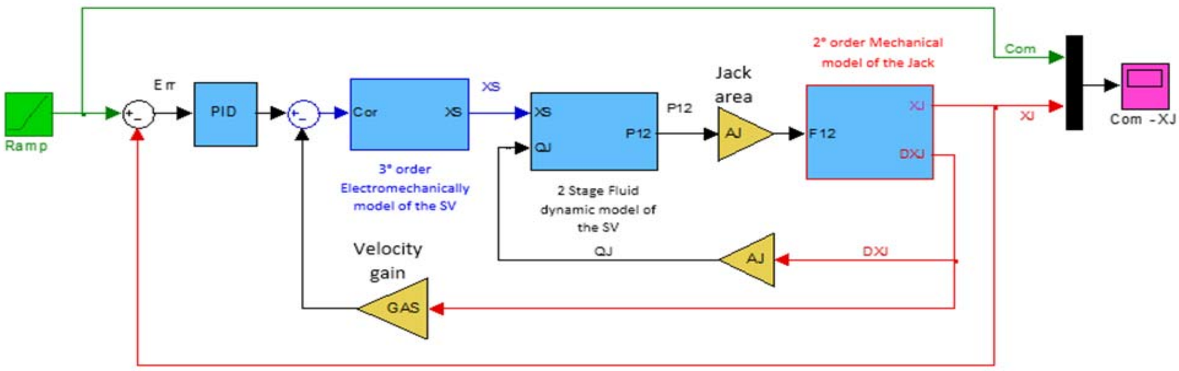

Figure 6. Ilustrates the MATLAB/Simulink block diagram representing the hydraulic system

2.5 Numerical model proposed as a first model

It explicitly refers to the Coulomb formulation, elucidating the influence of dry friction. This formulation encapsulates these effects, on the speed (v) and the active force (Fatt), as delineated in the modeling depicted in Figure 7.

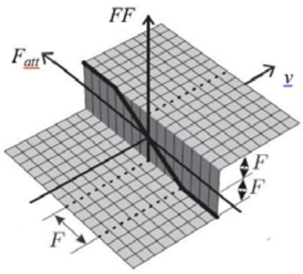

Figure 7. Representation of the proposed Coulomb friction model

The corresponding mathematical model is expressed by Equation 4:

$$FF = \begin{cases} F_{att} & if\ v = 0 \wedge |F_{att}| \le FSJ \\ FSJ \cdot sgn(F_{att}) & if\ v = 0 \wedge |F_{att}| > FSJ \\ FDJ \cdot sgn(v) & if\ v \neq 0 \end{cases} \tag{4}$$

Here, (FF) denotes the calculated friction force, (FSJ) represents the value of the friction force in static or adherent conditions, (FDJ) is the friction force in dynamic conditions, and (F_{att}) denotes the active force applied to the system.
To ensure numerical stability in the simulation programs based on mathematical model (4), precautions are taken to halt the mechanical system in the event of speed reversal during integration steps by:

$$v(t_{i+1}) = 0 \quad if \quad v(t_{i+1}) \cdot v(t_i) \le 0 \tag{5}$$

If the cessation of the system is not rectified, arising from an imbalance caused by an excess of active forces (F_{att}) in comparison to passive forces (friction), the subsequent calculation step could generate further discrepancies in the overall forces acting on the system, leading to a proper system restart. It is noteworthy to emphasize that the condition articulated in Equation (5) stands as a fundamental distinction and innovation of the current model in comparison to those documented in the literature [3]. The apparent obviousness of this condition underscores the robustness and precision of the method.

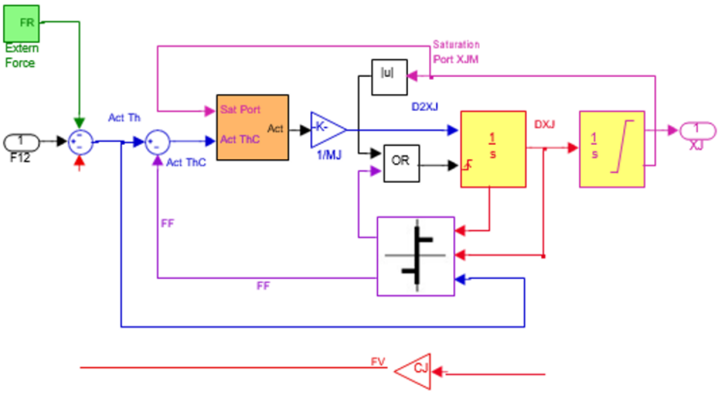

Figure 8. Depicts the MATLAB-Simulink block diagram representing the actuator model, incorporating dry friction and mechanical limit switches.

2.5.1 Coulomb friction model load-sensitive (FR). Building upon the earlier presented Coulomb friction model, a novel numerical simulation model has been devised. This model takes into account the friction forces, or momentum, as a function of the load (*FR*) on the user. In this formulation, the friction momentum is perceived as the sum of two distinct terms. The cumulative value of this sum, contingent on dynamic conditions, can adopt two distinct values, corresponding to either static or dynamic conditions:

The model computes the friction momentum through two components:

1. A term independent of the external load *FR*.
2. A term that is proportional to the load, determined by specified efficiency values.

The friction momentum calculation follows this formulation:

$$T_{FR} = T_{FR0} + \left(\frac{1}{\eta_{opp}} - 1\right) \cdot T_{LD} \tag{6}$$

In situations where the load opposes the motion.

$$T_{FR} = T_{FR0} + (1 - \eta_{aid}) \cdot T_{LD} \tag{7}$$

When the load conditions favor motion.

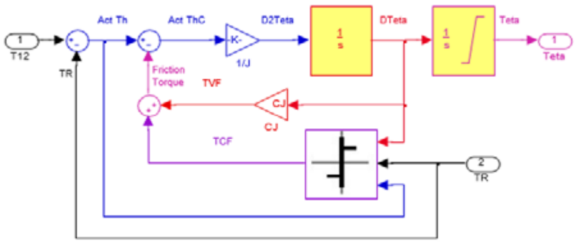

Figure 9. Simulink block diagram of the Coulomb friction model load sensitive

Subsequently, this mathematical model was implemented in the MATLAB-Simulink environment and integrated into a more intricate computational algorithm, giving rise to the numerical simulation program depicted in Figure 9 [9-12].

The comparison of these two models is conducted against two conventional calculation routines, each based on the assumptions outlined below.

Sign-function model "Discontinuous": The friction force is zero under conditions of rest and opposes the speed, remaining invariant in magnitude and equal to its dynamic value [11].

Continuous "hyperviscous" model with saturation: This model assumes a continuous and hyper-viscous nature for the friction force, featuring saturation characteristics.

The friction force, opposite to the speed, is proportional to it according to a high viscous coefficient (compatibly with computational instability) and is also limited in both directions to its dynamic value [7].

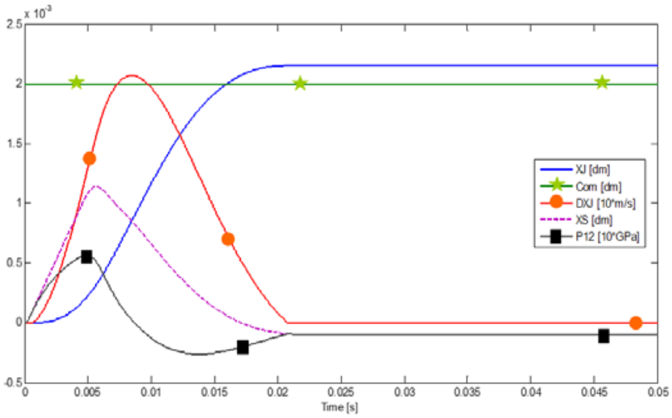

Figure 10. Step command with initial position 0 and final command 0.2 mm, with proposed Coulomb Friction model.

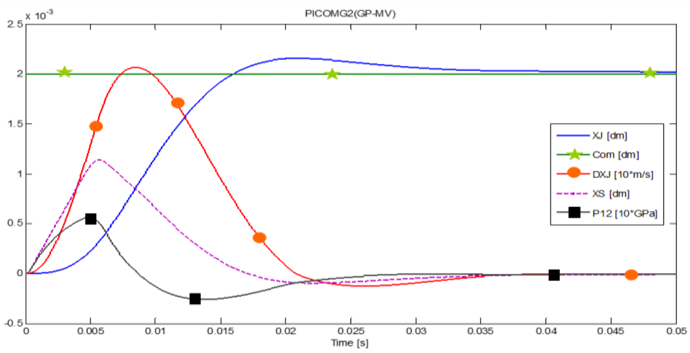

Figure 11. Step command with initial position 0 and final command 0.2 mm, with Continuous "hyperviscous" model.

3 Results

In Figure 10, a step command is illustrated with an initial position of 0 and a final command of 0.2 mm. The flow gain (GQS) is set at 0.2 m²/s, employing a purely proportional control logic. The scenario assumes the absence of hysteresis, leakage, and external loads acting on the jack, with a constant supply pressure.

At Time = 0, a command (Com) of 0.2 mm is initiated, leading to a position error since the commanded element has not yet transitioned from its initial position (XJ=0).

Because the inaccuracy in question is not significant, the servovalve undergoes a second stage (XS) displacement that does not include its limit switch. The XS position is counter-reacted at the first stage by the internal feedback spring between the two stages, thus generating a tendentially re-centering action towards the first stage, with consequent zeroing of the speed with which the second stage moves. The initiation of the second stage triggers the actuation speed (*DXJ*) of the jack, which is nearly proportional to *XS* (albeit slightly delayed). This relationship is established due to the low inertia of the jack and the absence of external loads. With reasonable accuracy, it can be asserted that the flow passing through the valve, which is proportional to *DXJ*, is also proportional to the displacement (*XS*) of the spool. The consequent inversion of the sign of the position error, at about Time=0.021 s, causes a new displacement of the second stage of the valve which would cause the inversion of the DXJ speed of the jack, if the dry friction were not able to keep it stationary despite the non-zero differential pressure due to the residual position error. It can be concluded that the system manifests a persistent position error, despite the absence of load on the jack and of constructive imperfections, which is exclusively attributable to the effects of dry friction model.

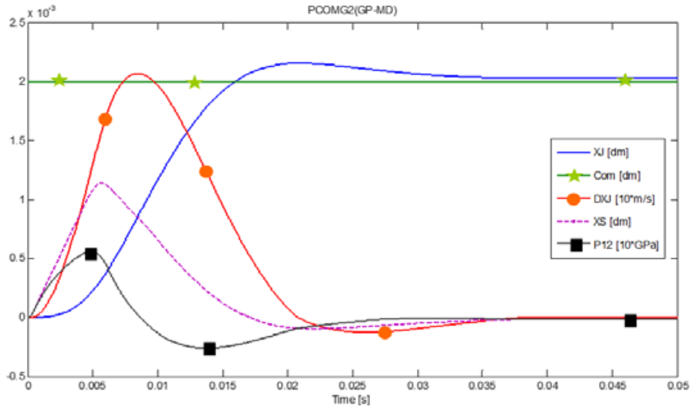

Figure 12. Step command with initial position 0 and final command 0.2 mm, with discontinues friction model.

The two other cases examined are completely identical to PCOMG2 (Fig. 10), but employ two different friction models widely used in the simulation programs: "hyperviscous" friction model with saturation PCOMG2 (GP-MV) Fig. 11 and “discontinuous” PCOMG2 (GP-MD) Fig. 12. In the dynamic response, no noticeable distinctions are observed until the overshoot peak is reached around Time = 0.021 s. Following this point, both models fail to accurately replicate the condition of rest (0DXJ=0) observed in PCOMG2. Instead, they exhibit a gradual convergence towards the commanded position. This slow convergence is marked by a sequence of speed reversals at each calculation step, characteristic of numerical instability, particularly in the case of PCOMG2 (GP-MD) due to its intrinsic discontinuity in the link between friction force and speed. If an excessively high viscous constant is chosen, a similar instability may be observed in PCOMG2 (GP-MV).

The inadequacy of the latter two models, "discontinuous" and "hyperviscous," is evident in their inability to accurately simulate the stationary condition with a persistent and constant position error.

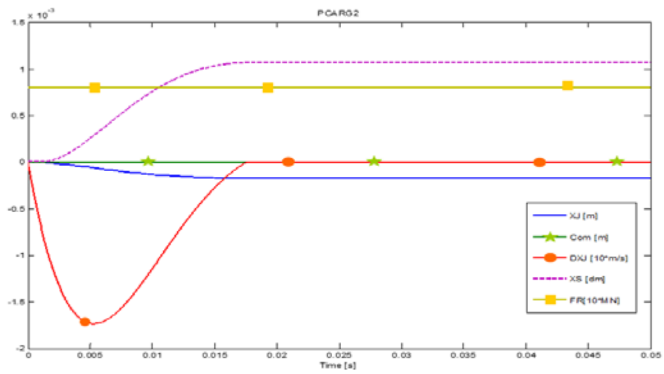

Figure 13. Step load with zero initial and final value of 8000 N with proposed Coulomb Friction model.

In the case of a step load with an initial and final value of 8000 N and a constant command of 0 m without actuation request, a disturbance is introduced by the load (Fig. 13). The flow gain (*GQS*) is set at 0.2 m²/s, employing a purely proportional control logic. The scenario assumes the absence of hysteresis and leakage, with a constant supply pressure.

At Time = 0, an 8000 N load is applied, inducing an un-commanded movement of the controlled surface. The dynamic behavior of the system correctly transitions towards a state of rest characterized by a persistent position error. This error is a consequence of the constant load (*FR*), to which a defined value of static friction force must be added based on stopping conditions.

It is important to note that the position error in stationary conditions would differ from the simulation result if friction were absent.

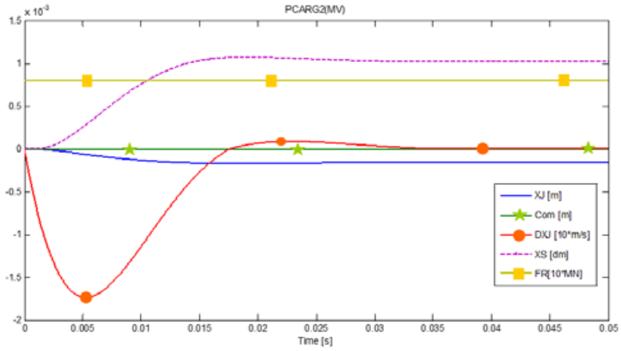

Figure 14. Step load with zero initial and final value of 8000 N with Continuous "hyperviscous" model.

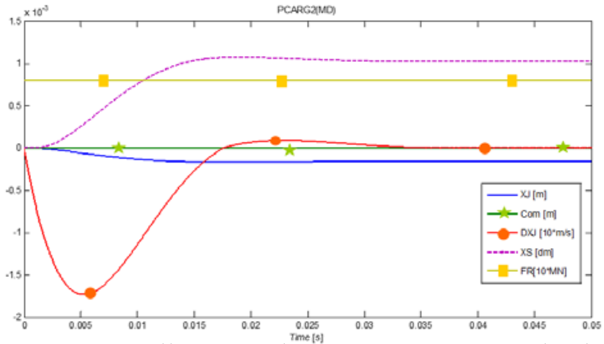

Figure 15. Illustrates the response to a step load with an initial and final value of 8000 N, utilizing the discontinuous friction model.

The two additional cases examined mirror PCARG2 (Fig. 13) but utilize two distinct friction models: the "hyperviscous" friction model with saturation, PCARG2 (MV) (Fig. 14), and the "discontinuous" friction model, PCARG2 (MD) (Fig. 15).

Similar to the previous scenario, no significant differences are observed in dynamic behavior until the overshoot peak is reached at around Time = 0.017 s. Subsequently, both models fail to accurately reproduce the correct rest condition (*DXJ*=0) of the controlled member as obtained in Fig. 13. Instead, they exhibit slow convergence, explained by a sequence of speed reversals indicative of numerical instability. This phenomenon is evident in PCARG2 (MD) (Fig. 15) and, only when an excessively high viscous constant is chosen, in the case of PCARG2 (MV) (Fig. 14).

The inability of the latter two models, "discontinuous" and "hyperviscous," to faithfully simulate the stationary condition with a persistently constant position error from the moment of overshooting is apparent.

4. Conclusions

The Coulomb friction model introduced in this study showcases its suitability for integration into computational programs aimed at simulating servomechanisms. It strikes a well-balanced compromise between modeling accuracy and computational efficiency, resulting in a compact algorithm.

The tests conducted reveal the efficacy of the proposed Simulink model, demonstrating its superiority over the built-in library routines provided by the Simulink program itself.

By incorporating the examined model alongside the viscous friction component in the speed feedback loop of the actuator, the fidelity of the numerical model for the entire servomechanism is significantly enhanced. This enhancement enables the simulation of various commands with satisfactory accuracy, addressing issues such as relative delays in starting, positioning errors, and responses under load, commonly associated with the influence of dry friction.

Used Symbols

XS	Second stage valve position	[mm/10] or [dm]
XF	First stage valve position	[mm/10]
Com	Command	[dm] or [degree]
PSR	Differential pressure controlled pressure	[MPA]
Teta	Position	[degree]
Dteta	Radial velocity	[100*rad/s]

DXJ	Velocity	[10*m/s]
P12	Differential pressure	[10*GPa]
XJ	Response of the system	[dm]
FR	Load	[10*MN]

* Other symbols and abbreviations, are explained in the paper.

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