Study of new fluid dynamic nonlinear servovalve numerical models for aerospace applications

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Abstract— **The development of modern flight control system requires more and more highly detailed computer models in order to analyze their specific behavior as a whole or related to their components and subsystems; on the other hand, especially in the phases of preliminary design or development of diagnostic or prognostic algorithms, it is necessary to implement simplified models that are able to combine sufficient levels of accuracy and reliability with reduced calculation times and computational costs. In this paper, the authors analyze the possibility of building more simple, synthetic models, with the aim to maintain an acceptable level of accuracy but being able to analyze the dynamic behavior of entire systems. This work focuses on the numerical models related to fluid-dynamics of servovalves, where the related algorithms are characterized by a semi-empirical formulation and will take into account the effects of variable supply pressure and leakage (which is related to the control ports connecting the valve to the motor elements). Two new models are proposed and compared with a detailed reference. This comparison is performed by evaluating the performance of the different models and their ability to describe the fluid dynamic behavior of the considered valve.**

Keywords-fluiddynamic; hydraulic; servovalve; simulation; simplified numerical model

I. INTRODUCTION

In actual modern civil and military aircrafts the flight control systems are becoming increasingly more complex but, according to highly severe standards in the aviation field, these control systems also must have rigorous requirements related to their performance and especially their safety; therefore, the design of these systems involves the use of highly detailed models for analyzing their behavior (at system, subsystem or component level). The introduction of more simplified and synthetic models is beneficial for dynamic simulation purposes of entire control systems, but they must guarantee suitable performances (particularly in cases of high workload regimes in regards of computer analysis of monitoring duties of the systems).

In other words, especially in phases of preliminary design or development of real-time (or near real-time) diagnostic or prognostic algorithms, it is necessary to implement simplified models that are able to combine sufficient levels of accuracy and reliability with reduced calculation times and computational costs. These synthetic models are especially suited to the systems monitoring, usually used both on the ground and in flight operations.

In fact, given that these functions must be performed in real time, they can require a high computational burden for the aircraft onboard processors and so, the implementation of lighter algorithms may result beneficial.

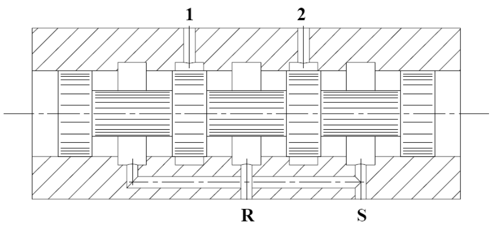

Figure 1. Schematic of the considered four ways valve

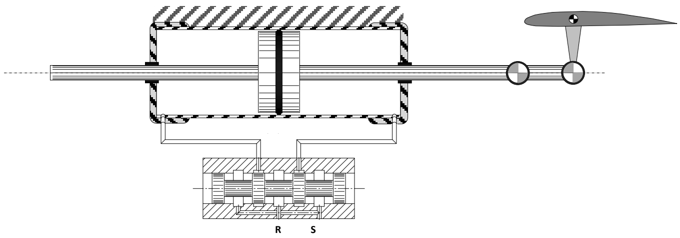

Figure 2. Schematic of the considered onboard hydraulic actuator.

For instance, these may be applied to the basic component of any proportional hydraulic control system, that are control valves, servovalves, actuators, etc.

II. SYNTHETIC FLUID-DYNAMIC MODELING OF VALVES

As shown in Fig. 1, the present work refers to a four-ways type control valve (supply port S, return port R, control port 1, control port 2), coupled with a linear jack through the two control ports (Fig. 2). The valve spool displacement XS rules the opening/closing of the four passageways, characterized by their overlaps/underlaps and connecting each control port to the supply and return ports respectively, so providing the desired relationship between flow and absolute pressure concerning each control port (P1 and P2), under defined oil characteristics [1-2]; the corresponding differential pressure, regulated between the two control ports is named P12. In zero-flow conditions, each control port absolute pressure is close to the supply one when the related passageway is much more open than the return one and particularly for all the spool displacement values performing a return passageway fully closed (saturation condition).

In the opposite case, the control port absolute pressure is close to the return one; in intermediate conditions, the control port pressures acquire medium amounts, having a progressive evolution between return pressure (PR) and supply pressure (PS) values, as it can be seen in the valve characteristic P12-XS of Fig. 3.

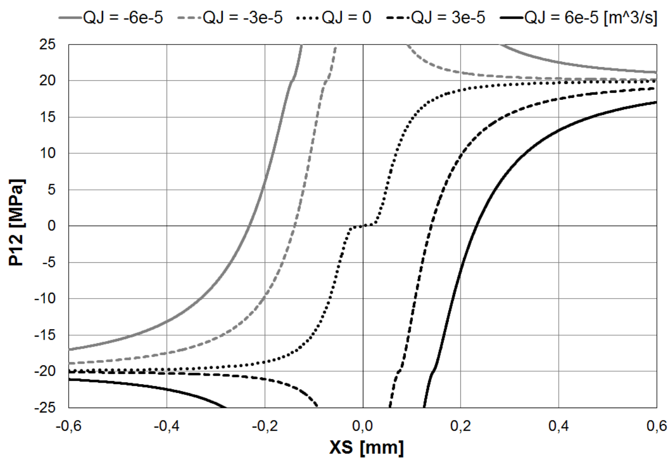

Figure 3. Characteristic P12-XS of the valve (HD fluid dynamic model)

It must be noted that the said valve characteristic P12-XS is a graph representing the evolution of the differential pressure P12 regulated by the valve as a function of the spool displacement XS and parameterized in the flow QJ provided to the user for a given valve geometry). Under non-zero flow, the control ports pressures may be markedly different with respect to zero flow conditions, particularly in case of small spool displacement; this behavior is the consequence of the oil flow QJ, forcing across the valve passageways. Figure 3 has been simulated by means of a detailed numerical model (HD) formerly proposed by Jacazio et al. in [3-4] and, subsequently, enhanced by Borello et al. [5]. The accuracy of this model was verified by comparing its outputs with the experimental results obtained by Urata [6-12] (as regards the servovalve electromechanical modelling), whereas the valve fluid dynamic model has been validated by certified numerical codes (e.g. Amesim or CFD softwares) and experimental data [13-17].

Simplified Fluid Dynamic Models in Literature

Any fluid dynamic control valve model should be able to take correctly into account the aforesaid considerations, but only an accurate approach, not explicitly dealt with in this work, is generally proper to describe the whole reported behavior, discriminating flow and absolute pressure affecting each control port [14]; when simpler and quicker models are requested or desired, as it is envisaged in the present work, only the controlled differential pressure between the two control ports P12 and a single flow value QJ (common to both control ports) are usually computed. Basically exist two main categories of fluid-dynamic control valve numerical models: a first type, suitable for regulating the controlled differential pressure acting on the motor element (e.g. linear actuators or hydraulic motors) and the other controlling the regulated flow in output [18]. As reported in [19], the former type describes the relationship between an output variable (i.e. the differential pressure imposed on the motor element) and an input variable (i.e. the commanded valve spool displacement), having as the feedback action the controlled flow through the motor element itself. As these categories of models have a multi-purpose generalist nature the aforesaid considerations are mostly valid in several cases, as the detailed complex model and the simplified ones considered in this work. The second category of fluid dynamic models, that will be not considered in this paper, gives the controlled flow as the output variable and uses the differential pressure as a feedback input, while the spool displacement is still assumed as the control input. In general, as regards the valve simplified numerical models available in the literature, the fluid-dynamic behavior is often simulated through a linearized approach based on two coefficients, easily obtainable experimentally, defined respectively pressure gain (GP) and flow gain (GQ) [20]. Therefore, adopting the said linear approach, it is possible to conceive the most simplified numerical model shown in Fig. 4: the spool displacement XS, through the valve pressure gain GP, produces a proportional value of differential pressure P12, which act on the motor element; this pressure is reduced by the pressure losses which are related to the controlled flow passing through the pressure/flow gain ratio GPQ. The most important weakness of this modeling is constituted to its impossibility to accurately simulate the effects of the supply pressure limits and, consequently, to calculate the actual stall conditions of the motor element. It must be noted that this linearized numerical model (Fig. 4) is not able to take into account the effects due to the maximum value of differential pressure PSR provided by the hydraulic supply or to an eventual pressure supply drops (e.g. a partial depressurization of the hydraulic system). To this purpose, a possible variant (derived from the previously described model) is reported in Fig. 5: it consists of the implementation of a saturation block acting on the differential pressure developed through the related gain. In this way, it is possible to progressively enhance the model performance computing the effects of the differential supply pressure PSR. It must be noted that the so described simple measure has a severe shortcoming, underestimating the actuation rate in case of a fully open valve: this is particularly noticeable when a the valve reaches the saturation condition for small spool strokes (with respect to the maximum spool travel XSM).

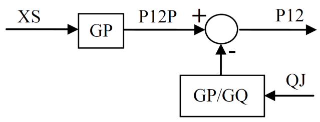

Figure 4. Linearized numerical model of the valve fluid dynamics

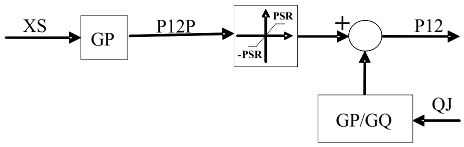

Figure 5. Nonlinear numerical model of the valve fluid dynamics

In a previous work [19], the authors proposed four new simplified numerical models (A, B and C-type briefly shown in the next section) derived to the ones described here above: they have been conceived to better simulate the differential pressure limits which are connected to the hydraulic power supply. However, all these models put in evidence, albeit to varying degrees, the same limits, essentially related to leakage and variable pressure supply (as reported hereafter). Moreover, as regards the C-type models, the simulation of the aforesaid leakages determines an instantaneous feedback loop that can generate computational problems.

Simplified fluid dynamic models A, B, C1 and C2

The approach proposed by the authors in [19] takes into account the effects of the variable values of the supply/return differential pressure performed by the hydraulic system, (reported as PSR): to this purpose, it must be noted that the effect of PSR on both pressure (GP) and flow (GQ) gain amounts can be computed in a reasonable but simple form by considering the general layout of the models, which is as linear as possible and, consequently, is conceived around the acceptable hypothesis of a linear relationship between each considered gain and the value of PSR. The actual values of GP are sufficiently close to the proportionality with respect to PSR, but the actual values of GQ are more and more close to the proportional to the square root of PSR. In this situation, it can be assumed that the proportionality between the GQ and the PSR it is not so realistic, but it will be accepted in our models as it is in line with the proposed linearized approach. Therefore, the pressure to flow gain ratio (GP/GQ) can be assumed as independent on the value of PSR. In the same way, also the value of the spool displacement XS at which the differential pressure P12, produced on the motor element in the zero-flow condition (defined P12P), saturates to PSR can be considered independent on PSR; this critical spool displacement (hereafter reported as XSS) is characteristic of the different types of valves and dependent on the internal geometry of the spool. According to these assumptions, the value of P12P can be computed dividing XS by XSS and multiplying it by the value assumed by PSR in the present situation, as shown in Fig. 9; further, the pressure to flow gain ratio GP/GQ can be replaced by the coefficient GPQ, which is invariant with respect to the supply differential pressure

PSR. As regards the leakage model and related computational algorithm, the authors introduced the following considerations. The aforesaid leakage is modeled as the sum of all the fluid-dynamic losses that determine oil flows through the sealing elements of the valve spool. In other words, the flow controlled by the valve passageways, and driven to the ports 1 and 2 (Fig. 1), mainly operates across and within the motor element, displacing a proper fluid volume and developing mechanical power, but, usually, a small amount of it flows, across imperfect seals or intentional bypass devices (based on calibrates orifices), directly from port 1 to 2 or vice versa (so being unable to perform any useful work). Nevertheless, it produces further pressure losses across the valve passageways, besides those developed by the operating flow. Also in this case it is reasonable to assume a linear dependence between the differential pressure P12 and the leakage flow drained (QLk); this simplified hypothesis is generally admissible as it is assumed that leakage generates small oil amounts flowing through very small dimension passages. So, the relationship between P12 and QLk should be expressed as:

$$\mathrm{QLk} = \mathrm{CLk} \cdot \mathrm{P12} \qquad (1)$$

where the CLk is the leakage coefficient. As depicted in Fig. 6, the valve leakage can be modeled by the feedback loop containing the CLk block; it must be noted that CLk represents the ratio between the leakage flow QLk and the differential pressure P12 (which generates this fluid loss).

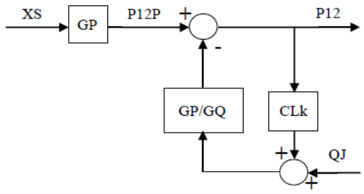

Figure 6. Linear fluid dynamic model sensitive to valve leakage

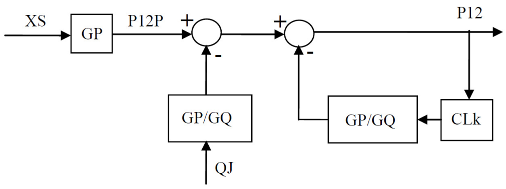

Figure 7. Alternative version of Fig.6 by separating QJ and CLk loops

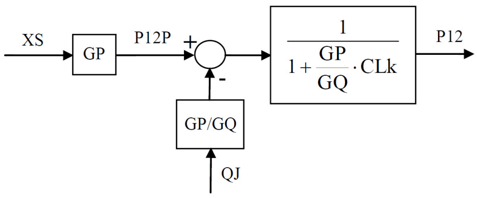

Figure 8. Valve leakage loop solution

The total oil flow disposing through the valve control passageways should be calculated summing the leakage flow QLk and the controlled working flow QJ; also it can obtain the related differential pressure loss multiplying it by the pressure/flow gain ratio GPQ. It must be noted that the computational structure shown in Fig. 7 is afflicted to a meaningful numerical shortcoming: the leakage feedback branch, containing only algebraic blocks, constitutes an instantaneous loop causing numerical instabilities.

This problem is overcome rewriting the computational algorithm by a different formulation based on the preventive analytical solution of the said leakage instantaneous loop (Fig. 8). Starting from the block diagram shown in Fig. 8, authors proposed in [19] a development of the said linear fluid dynamic model (Fig. 6), which takes into account the effects of leakage and variable supply differential pressure PSR: this model, named MODEL A, is shown in Fig. 9. Taking into consideration the same effects of the variable value of PSR as well the leakage, it is possible to develop the nonlinear model represented in Fig. 2, obtaining another configuration, named MODEL B (Fig. 10).

However, as reported in [19], this modeling, although more complex than the previous one, is completely unsatisfactory.

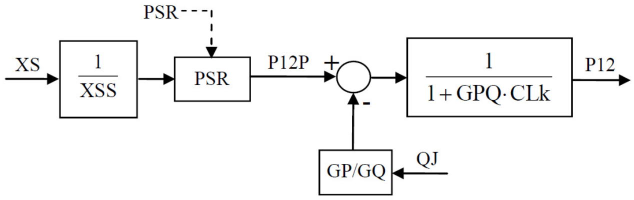

Figure 9. MOD. A: linear valve model sensitive to PSR and CLk [19]

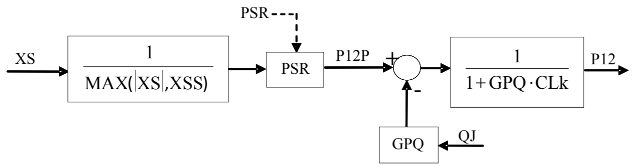

Figure 10. MOD. B: non-linear model sensitive to PSR and CLk [19]

A modified version of the nonlinear numerical model shown in Fig. 5 have been proposed in [20]; the main difference regards the position of the pressure saturation block which, in this alternative case, is positioned downstream the flow feedback, as it can be seen in Fig. 11. The advantage offered by this layout consists of the ability to take acceptably into account the effects of P12 limitations on the actuation rate, so obtaining a more proper value of the no-load actuation rate itself. On the contrary, the shortcoming of this model is represented by the inability to simulate the temporary overload conditions, eventually affecting the motor element. In this layout, the weak evaluation of overload conditions is generally not considered so important, and, vice-versa, the better performance in evaluating the motor actuation speed (providing in this way a more precise value of the no-load actuation rate) is significantly considered. Several models/algorithms has been developed starting from the block diagram of Fig. 11 to analyze the fluid dynamic behavior of a given valve taking into account the effects due to leakage and differential supply pressure PSR [19-20]. The main goal of these nonlinear models was combining two opposite (and often antithetical) characteristics: the maximum simplicity to represent the physical system (i.e. reduced computational burdens) and the required high accuracy (i.e. its fidelity in simulating the actual fluid dynamic behavior). A first model derived from the scheme of Fig. 11, named MODEL C1 in [19], is shown in Fig.12. It includes leakage and variable PSR computational algorithms and the flow feedback sum block, nevertheless being upstream the saturation block PSR, has been displaced downstream the GP one in order to use the invariant GPQ block. In this way, the leakage feedback loop is fully located downstream the pressure saturation block, limited within the values ± PSR. MODEL C1 must be able to simulate the effect of variable values of PSR along the simulation run, evaluating (according to the above-discussed assumption of proportionality between GP, GQ and PSR) the relative variable values of pressure and flow gains. Furthermore, the leakage feedback loop must be previously analytically solved, to avoid problems of computational instability.

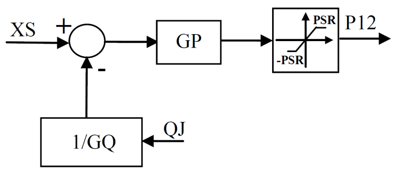

Figure 11. Alternative nonlinear model of the valve fluid dynamics

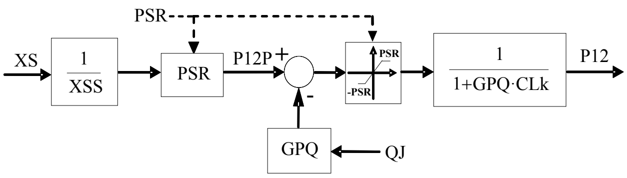

Figure 12. MODEL C1 [19]

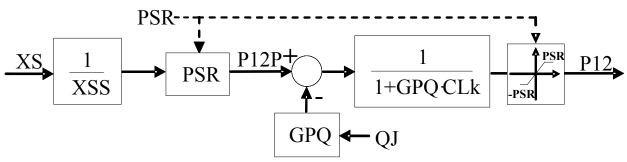

Figure 13. MODEL C2 [19]

Another model derived from the scheme of Fig. 11, named MODEL C2 in [19], is shown in Fig.13. In this formulation the leakage loop is entirely located upstream the pressure saturation block, limited within the values ± PSR. As widely described by the authors in [19], the major drawback of both MODEL C1 and C2 is related to the pressure gain contained in GPQ coefficient that, not taking into account the differential pressure saturation, could generate problems about the evaluation of leakage effects.

III. PROPOSED SIMPLIFIED FLUID DYNAMIC MODELS

In this paper, the authors will propose some new synthetic formulations which are intended to enhance the behaviors of the previous C-type models [19], taking into account the effects of variable supply pressure and leakage acting among the control ports connecting the valve to the motor element (i.e. the eventual PSR variation, from a computational point of view, affects both pressure and flow gains, besides the direct action on the pressure limits).
As reported in the previous chapter, as a consequence of the leakage feedback loop these models could be suffering from numerical instabilities and other computational problems; for this purpose, as explained hereafter, the authors propose two possible solutions (i.e. the new C3 and C4 models). The first enhancement proposed by the authors in order to overcome the limits evidenced by the previous models is shown in Fig. 14: to this purpose, it is proposed to overcome the problems related to the interaction between the pressure saturation block and the leakage feedback loop (located downstream of this block) by modifying the formulation of the pressure/flow gain ratio GPQ. As already highlighted in [19-20], even if the GPQ value is almost independent of the differential supply pressure (PSR), the effect of leakage on the regulated pressure downstream of the valve (P12) is less significant for large spool displacements (when |XS| > XSS).

This can be explained by remembering that, with high spool displacements, the control ports areas (regulating the actuation flow QJ) are much greater than internal valve orifices (through which the said leakage flows are drained).

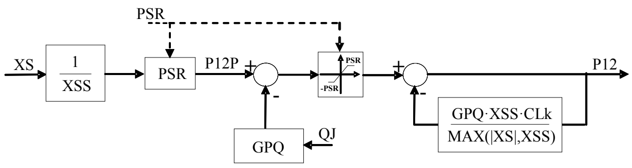

Figure 14. MODEL C3 initial formulation

As reported in [14], under linear conditions the pressure gain GP is almost independent of the spool displacement XS and, therefore, the regulated differential pressure P12 should be calculated as follows:

$$\mathrm{P12} = \mathrm{GP}\cdot\mathrm{XS} \qquad (2)$$

Vice versa, under saturation conditions (i.e. |XS| > XSS), (2) is no longer valid, and the apparent value of conditions the pressure gain GP(PSR,XS) (i.e. related to XS and PSR) decreases progressively as XS increases: therefore, in this case P12 should be calculated as:

$$\mathrm{P12} = \mathrm{GP(PSR,XS)}\cdot\mathrm{XS} \qquad (3)$$

Taking also into account that, in condition of pressure saturation (i.e. |XS| = XSS), P12 is equal to PSR, it is possible to express GP as follows:

$$\mathrm{GP} = \mathrm{PSR/XSS} \qquad (4)$$

To develop a general formulation of GP(PSR,XS) shown in (3) (and defined GPSS in the following), valid both for linear and saturated conditions, (4) should be modified as:

$$\mathrm{GPSS} = \mathrm{PSR/MAX(|XS|,XSS)} \qquad (5)$$

where MAX(|XS|,XSS) represents the Matlab function "maximum", which calculates the highest value between | XS | and XSS. Thus, by combining (4) and (5), a new GPSS formulation is obtained depends on the linear GP, the spool displacement XS, and the XSS:

$$\mathrm{GPSS} = \mathrm{GP\cdot XSS/MAX(|XS|,XSS)} \qquad (6)$$

The enhanced pressure gain formulation proposed in (6) is implemented in the leakage feedback loop of MODEL C3 (as shown in Fig. 14) in order to mitigate the shortcomings highlighted in the last section of the previous chapter. For this purpose, the pressure to flow gain ratio GPQ (i.e. GP/GQ) adopted in previous models has been modified according to (6): the overall gain of the leakage feedback loop (formerly equal to CLk·GPQ = CLk·GP/GQ) is then modified substituting the constant pressure gain GP (independent to XS) with the proposed GPSS, so obtaining:

$$CLk \cdot GPSS/GQ = CLk \cdot GPQ \cdot XSS/MAX(|XS|,XSS) \quad (7)$$

Also in this case, as has already been done for the first two C-type models, it is possible to pre-resolve the leakage feedback loop obtaining the final formulation shown in Fig. 15 as MODEL C3.

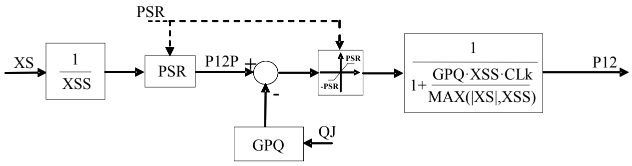

Figure 15. MODEL C3 final formulation

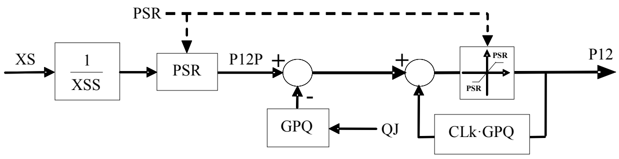

Figure 16. MODEL C4 initial formulation

A further possible development of the above mentioned C-type models [19], including leakage and variable PSR computational algorithms, is here introduced by the authors. as MODEL C4 (Fig. 16). In this model, in order to employ the invariant GPQ block, the flow feedback sum block has been displaced downstream the GP one, nevertheless being upstream the saturation block PSR. To compute the effects of the variable value of PSR, the model must be compliant not only with variable values of PSR along the simulation run, but also with the related variable values of pressure and flow gains, according to the above discussed assumption of proportionality between GP, GQ and PSR. Theoretically, given that MODEL C4 considers the leakage loop including the pressure saturation block (limited within the values ±PSR), the authors expect this formulation results as more realistic than MODELS C1, C2, C3. It should be noted that this computational layout is not necessarily consistent with the previous analytical solution of the loop itself (as proposed for instance in Fig. 15), that is able to prevent computational instabilities, and so, MODEL C4 could generate transitory numerical troubles. Consequently, as regards this model, a different solution of the problem is considered: to preventing any instantaneous dynamics, the leakage loop is converted in a first-order subsystem characterized, for example, by a hydraulic capacity. By giving a proper value to the related hydraulic time constant τ (consistent with the integration step DT adopted in the numerical simulation code), these computational instabilities can be avoided, but the dynamic behavior of the whole system may be improperly modified. At the beginning of each new calculation step of the numerical algorithm simulating the MODEL C4 response, a brief simulation of the dynamics of the leakage loop is iteratively run, until the reasonable stabilization of its pressure and flow values. This iterative method is based on the first-order pseudo-dynamic approach proposed by Borello et al. in [21-22]. As a consequence, the final evolution of this model, similarly developed as the previous MODEL A, is reported in Fig. 17: in this figure, the "short run" of the leakage dynamics is dash-outlined with gray background.

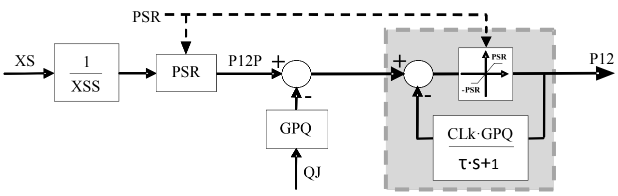

Figure 17. MODEL C4 final formulation

The merits or demerits of the aforesaid valve models, characterized by a semi-empirical formulation, are only related to their ability to properly describe the fluid-dynamic behavior of the valve, represented by the diagrams reporting their "characteristics" and by the simulations of a typical servomechanism employing it. The related considerations are presented in the following paragraph. To this purpose, some dedicated computational programs have been prepared.

IV. Numerical Results of EHA Testbench

In order to compare the behaviour of the different models and related computational algorithms concerning the fluid-dynamics of the control valve equipping a hydraulic actuation servomechanism, a typical system was considered. The conceptual schematic of this electro-hydraulic actuator (EHA) system is shown in Fig. 18. It mainly consists of a Power Control and Drive Unit (PCDU) and its control is performed by an Electronic Control Unit (ECU), not shown in Fig. 18, closing the position control loop. The PCDU contains hydraulic piston, and control electro-hydraulic two stage servo-valve. The EHA model takes into account the electrical, hydraulic and mechanical characteristics of all the system components which are relevant to the purpose.

In particular, the model is able to compute the following:

- inertia, viscous and eventual Coulomb friction regarding the hydraulic piston;
- hird order electromechanical dynamic model of the servo-valve with first and second stage ends of travel and simplified fluid-dynamic model, containing the motor element internal leakage.

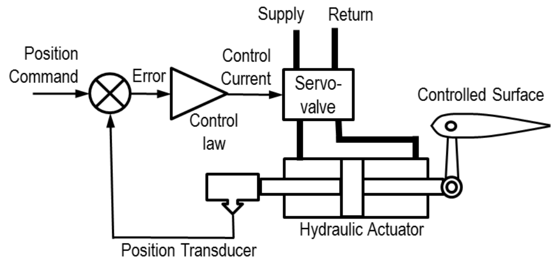

Figure 18. Schematic of the electrohydraulic actuator (EHA) [23]

The simulations shown in this chapter in Figs. 19 to 22 represent the dynamic response of the aforementioned EHA to a combination of position controls (Com), external loads (FR) and variations in the hydraulic supply pressure (PSR): this sequence of input has been appropriately defined to highlight the performance of the proposed fluid dynamic-models and their effect on the dynamic behavior of the EHA simulation testbench. These figures highlight differences, strengths and weaknesses of the simulation models obtained by implementing the fluid-dynamic model of the SV coil using the different algorithms proposed in the previous chapters: MODEL C1 (Fig 19), MODEL C2 (Fig. 20), MODEL C3 (Fig. 21) and MODEL C5 (Fig. 22). Fig. 23 shows the dynamic response of the same numerical simulation model of the servomechanism equipped with a high-fidelity valve fluid dynamics simulation model (HD MODEL); it is considered a reliable tool able to perform accurate simulations and, therefore, it is used as a reference to evaluate the said simplified models. According to [19], the time history applied to Com consists of a series of three step commands ranging from 0 m (initial position) to 0.02 m at Time =0 s, to 0.03 m at 0.3 s, to 0.02 m at 0.75 s. The time history of the load FR acting on the motor element, having null value since 0 s to 0.2 s, reaches the final constant value (10400 N) through a step change at Time = 0.2 s; so, the actuation run of the system following the first step command is unloaded, while FR acts as an opposing or aiding load during the second run (starting at Time = 0.3 s) or the third one (Time = 0.75 s and following) respectively. The time history of the supply/return differential pressure PSR consists of three time intervals, each characterized by a constant differential pressure value: during the first and the third time interval (Time since 0 s to 0.35 s and since 0.45 s to the end of simulation, respectively), the 20 MPa nominal value is kept as a constant (corresponding stall load FR = 14.1 kN), while during the second (0.35 to 0.45 s) time interval the constant 12 MPa reduced value (related stall load FR = 8.5 kN) is performed through two-step changes. So, the effect of a temporary supply pressure drop, acting during the opposing load actuation run, is evaluated. All these simulations have been run with a leakage coefficient $CLk=2\cdot 10^{-13}$ m3/s/Pa. Figs. 19, 20 and 21 present the dynamic behavior of the system according to MODEL C1, MODEL C2, MODEL C3 respectively, in comparison with MODEL HD. Both the unloaded and the aiding load actuation runs are rather accurately simulated, in spite of lower stopping deceleration and slightly higher starting accelerations, as the P12 behavior proves. The opposing load actuation runs reveal some significant discrepancies with respect to MODEL HD: the load effect on the system actuation rate, in terms of reduction of the rate itself, is underestimated and, when the supply pressure drops, the system back movement is overestimated, performing an incorrect constant back acceleration. Further, the acceleration following a spool displacement change keeps a constant value along a relevant part of the acceleration transient, rather than the much more plausible asymptotic trend reported in MODEL HD, similar to a first order response which follows a step input; the reason lies in the simple but partially unsatisfying action of the P12 saturation block that is implemented within the algorithms represented by the block diagrams reported in Figs. 12, 13 and 15. In these conditions, the results given by MODEL C1, MODEL C2 and MODEL C3 are unreliable with respect to the surely more accurate MODEL HD ones (Fig. 23), but the computational inaccuracies ascribable to MODEL C2, MODEL C3, MODEL C1 are high, higher and much higher respectively and are emphasized by increasing Clk values in MODEL C3 and MODEL C1. Similar considerations regard the stop following the run in aiding load condition, performing an incorrectly delayed action.

It proves the inability of MODEL C1, MODEL C2 and MODEL C3 to take correctly into account the damping action related to the flow crossing the valve passageways, when load and deceleration require particularly high P12 values, eventually exceeding PSR. This improper behavior depends on the restrictions imposed on the simulated P12 pressure level, without regarding the specific working conditions of the system; in fact, MODEL C2 limits the P12 amount within ±PSR, whatever Clk value is, MODEL C3 and mainly MODEL C1 limit P12 within ±PSR*, where PSR * decreases more and more (compared to the nominal value of PSR), as the value of Clk increases (i.e. PSR* is to be intended as PSR reduced by leakage effect).

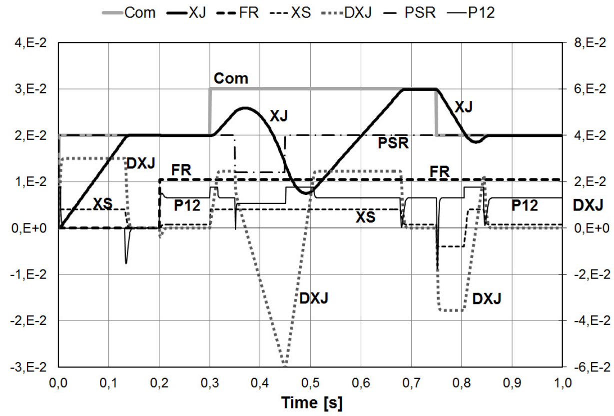

Figure 19. Dynamic response of the EHA model – MODEL C1

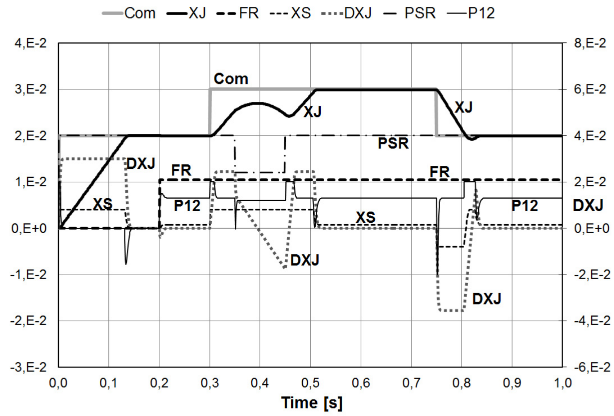

Figure 20. Dynamic response of the EHA model – MODEL C2

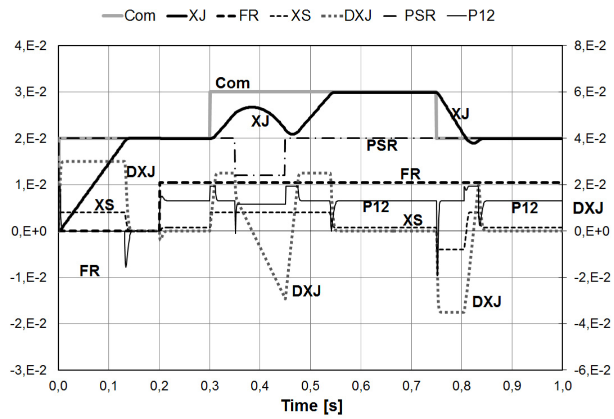

Figure 21. Dynamic response of the EHA model – MODEL C3

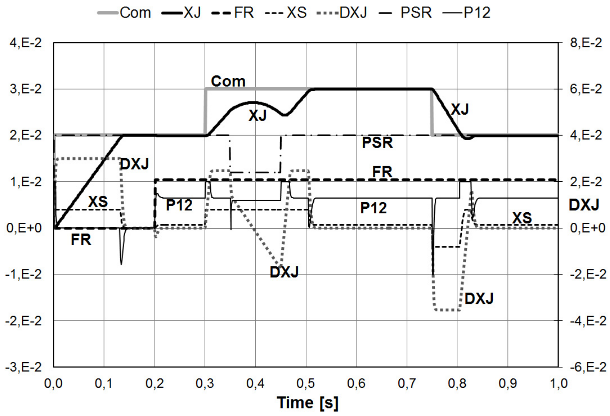

Figure 22. Dynamic response of the EHA model – MODEL C4

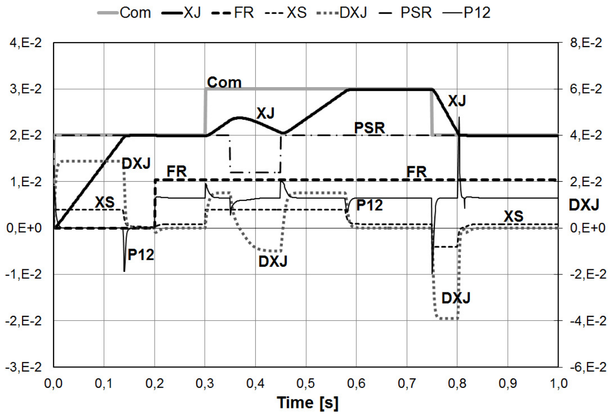

Figure 23. Dynamic response of the EHA model – HD MODEL

In case of an actual system having the valve spool fully displaced, the stall load characterizing the piston decreases as Clk grows and it is a valuable aspect of MODEL C1 and MODEL C3 with respect to MODEL C2, but, if an over-stall load is reached, the inability to perform properly high P12 levels represents a severe shortcoming, mainly for MODEL C1 and MODEL C3. Instead, regarding MODEL C4 (Fig. 34), it must be noted that, contrary to all authors expectations, also in this case it is not able to give any significant improvement with respect to the simpler MODEL C2 [19] (despite its greater complexity and possible numerical stability problems associated with the first-order model of leakage ring); the same results as MODEL C2 are ascribable to the concrete inability of the saturation block to have any influence upstream through the leakage feedback, because the pressure limits themselves cut off any effect.

V. Conclusion and Future Perspectives

The analysis of the performances of the five different fluid dynamic models of valve considered in this document (C1, C2, C3, C4 and HD) clearly highlights the few advantages and the shortcomings of the simplified models proposed so far by the authors. The evaluation of transients (accelerations, decelerations) is more or less deficient (over- or under-estimated) in all the models. The simulations of both the unloaded and aiding loaded actuation runs are sufficiently accurate in MODEL C1, MODEL C2, MODEL C3 and MODEL C4.

The computational evaluations of the actuation run in conditions of opposing load are generally unsatisfying, but particularly for MODEL C1 and MODEL C3, because of the overestimation of the actuation rate itself. The over-stall condition, when an actuation run is commanded in opposing load condition, produces a substantial overestimation of the actuation speed DXJ in all the proposed models (in comparison with the HD MODEL) but it is much more marked in the case of MODEL C3 and especially MODEL C1. In general, the proposed C-type models are not completely capable of overcoming the shortcomings of previous models. However, especially under saturation conditions, MODEL C2 appears to be sufficiently more accurate than others, particularly in the case of low QJ value, providing some small improvement with respect to all the other models here considered. Instead, as already mentioned, it should be noted that, contrary to all expectations, MODEL C4 is not able to make any substantial improvement compared to the C2 MODEL, despite the greater complexity, the higher computational cost and the possible problems of numerical convergence. In conclusion, the proposed approaches to the modeling of typical non-linear fluid dynamics, which characterize the proportional control valves, present some gaps, in particular in non-linear fields. The proposed new models, while trying to propose more efficient solutions, have not proved to be decisive. In the authors' opinion, further studies are needed, capable of producing more efficient algorithms, to improve the ability to perform acceptable simulations of all possible working conditions.

TABLE I. LIST OF SYMBOLS

Symbol	Definition	Units (SI)
CLk	Valve leakage coefficient	$m^3/(Pa\cdot s)$
Com	Servomechanism position command	m
DXJ	Motor element velocity	m/s
FR	Load acting on the motor element	N
GP	Valve pressure gain	Pa/m
GQ	Valve flow gain	m^2/s
GPQ	Pressure to flow gain ratio (GP/GQ)	$Pa\cdot s/m^3$
P12	Actual differential pressure	Pa
P12P	Zero-flow differential pressure	Pa
PSR	Supply/return differential pressure	Pa
QLk	Leakage flow	m^3/s
QJ	Working flow	m^3/s
XJ	Motor element position	m
XSM	Spool end of travel displacement	m
XSS	P12P saturation spool displacement	m

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