Interaction Problems Between Users in the Design of Hydraulic System

L. Borello, G. Villero, M. D. L. Dalla Vedova

***Abstract*—The design of the whole airplane hydraulic system should be seen in an integrated form: in fact different configuration choices regarding each subsystem can appreciably affect the pumps sizing. The work highlights the interactions between the design choices concerning both the generation and power regulation (pumps and regulating devices) and the users in all the aspects important for the operation of the complete system outlining the design of a defined hydraulic system of an airplane. To this end the dynamic simulation models (with associated calculation programs) capable of analyzing the behavior of the complete system have been made; employing it a series of investigations was carried out, aiming to analyze the dynamic behavior of different configurations of the system in similar operating conditions. Examination of the results shows how seemingly minor design choices concerning the architecture of the subsystems can play a significant role in the functioning of the entire system, affecting its own sizing.**

Keywords-aircraft hydraulic system, users interaction, power regulation

I. INTRODUCTION

The design of an airplane hydraulic system consists, firstly, in the sizing of its pumps and related devices for the supply pressure regulation. The design parameters are essentially obtained from the diagrams that express the workload of the subsystems connected to the system itself. The above diagrams take into account the flight portion in which each subsystem works, the maximum flow rate required in such conditions and the propulsion engine angular rate related to the rate of the pumps directly connected to the gear box. These diagrams give, in each flight portion, the maximum flow required by the subsystems and consequently requested to the system pumps. The latter, always of volumetric type as a consequence of the high prevalence required, provide a flow proportional to their displacement so as to their angular rate and slightly decreasing with the prevalence imposed by the system. From the previous considerations the minimum value of the displacement requested to the pump in order to properly move the assigned subsystems can be obtained. It should however be noted that, in conditions of reduced flow to the subsystems, a suitable device must adjust the supply pressure, otherwise doomed to grow very rapidly and intensely (volumetric pumps) with eventual detriment not only regarding the correct subsystems functionality but also the same system integrity.

In particular it can be noted that the system and subsystems design should be studied in an integrated form: in fact different configurations regarding each subsystem can greatly affect the pumps sizing. Suffice it, for example, to remind that in a flap command the reversibility or irreversibility of the final actuators or the control logic used appreciably affect the actuation rate, particularly in aiding load conditions (retraction, so absence of supplied mechanical power) with consequent different values of requested flow rate.

It is also convenient to consider some types of limitation in the contemporary operations of the subsystems in every flight condition; it can be performed studying appropriate actuation sequences, if these do not affect the safety levels of the aircraft mission.

II. AIM OF WORK

Aim of the work is to highlight the interactions between the design choices concerning both the generation and power regulation (pumps and regulating devices) and the subsystems in all the aspects significant for the operation of the whole system, both in the current state of art and in foreseeable future. It also aims to suggest alternative logics able to establish appropriate sequences of actuation depending on the considered subsystems and the flight condition criticality. To this purpose, for example, the authors consider the hypothetical design of an aircraft hydraulic system equipped with three subsystems of different characteristics and dimensions, as defined by a complete series of operational requirements [1-4]; in fact the analysis realistically starts from the only knowledge of specifications concerning the subsystems and critically analyzing different possible solutions for architecture and sizing of subsystems and power generation.

III. SYSTEM ARCHITECTURE

Typically, the mental process to be followed in the design of a hydraulic system starts from the requirements which must be met by the subsystems to be connected; this leads, after

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some initial assumptions, to the subsystems conceptual design that allows in turn defining the requirements which must be met by the hydraulic system. The following step is then the system itself conceptual design.

Subsequent refinements of the overall design lead to the detailed definition of the various subsystems controlling devices, affecting their performance in stationary conditions, their dynamic behaviors and their typical interactions. The analysis of these results allows partial or radical design changes in order to solve any problems that may characterize the adopted configuration.

These considerations have been applied to the design of a hydraulic system intended to supply one primary and two secondary flight controls; this choice is however considered, in its simplicity, significant of all the typical problems of interactions between subsystems that may arise in a system more realistically equipped with a high number of subsystems. In particular, the considered system is composed of a fixed or variable displacement pump (in the latter case controlled by a compensator) equipped with a supply pressure controlling and stabilizing device; this is based on an accumulator and on one or more pressure control valves, as well as by three subsystems consisting of position servos (proportional or on/off) with servovalve, cylinder or hydraulic motor, motion transmission and related position control loop (Fig.1).

In particular, the work analyses the case of a military transport aircraft equipped with:

- longitudinal primary hydraulic flight control (hereafter called user 1);
- secondary hydraulic flap control (user 2);
- rear loading ramp door hydraulic actuator (user 3)

performing an air-drop operation in which a simultaneous actuation of the three aforementioned users may be required.

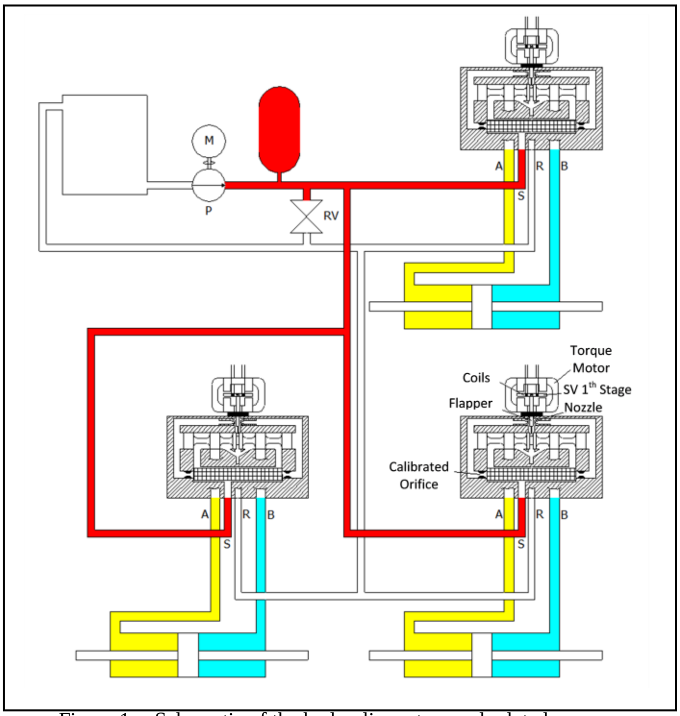

Figure 1. Schematic of the hydraulic system and related users.

IV. Mathematical Model and Simulation Program

The dynamic models, the corresponding mathematical algorithms and simulation programs can be represented in the form of a block diagram according to Fig. 2 as regards the user 1, according to Fig. 3 in respect of the users 2 and 3 and in accordance with the Fig. 4 with regard to the generation and control power system. As it can be seen from Fig. 2, the error resulting from the comparison between the commanded θC and effective θ_J positions is processed, as regards the user 1, through a logic of the proportional (G_{AP}) - integrative (G_{AI}) - derivative (G_{AD}) type equipped with a limiter at the integrator output (e_{IM}) to obtain the driving current through the servovalve first stage torque motor; this, by the torque motor gain G_M, provides the torque acting on the valve first stage which, net of the feedback effect (K_{SF}) coming from the second stage position X_S, gives the first stage position X_F (limited within the ends of travel X_{FM}) according to a second order dynamic characterized by the first stage elastic hinge stiffness K_F, by the circular frequency σ_{nF} and by its damping ratio ζ_F. The first stage position, operated through its flow gain G_{QF} and the area of the second stage end faces A_{SV}, gives the second stage speed which, by a time integration, identifies the position (limited within the ends of travel X_{SM}). From this, through the second stage pressure gain and the effects of the differential pressure saturation, is obtained, net of the pressure drops (depending on the ratio G_{PQ} between the valve second stage pressure to flow gains) due to the total flow Q_J requested by the hydraulic piston, the differential pressure P_{12} actually acting on the latter. It identifies the leakage flow through the proper coefficient C_{LK} and, by piston area A_J and mass M_J, net of load F_R, viscous (coefficient C_J) and dry friction [5-6], gives its acceleration ($d^2\theta_J/dt^2$); the latter, through a time integration, gives the speed ($d\theta_J/dt$) which defines the values of viscous friction, dry friction and jack working flow. The last, added to the leakage flow, gives the pressure drop through the valve passageways. The speed integration gives the actual position of the controlled element (θ_J) which is closed in loop on the command, so performing the position error value. With regard to user 2 and user 3 (Fig. 3) the same considerations apply with the exception of the control logic: in fact the current driving the servovalve is obtained by the position error through a proportional law with speed servo loop instead of a PID type, and is equipped with a ramp generator input command. As reported in Fig. 4, the flow rate Q_P supplied by the pump minus the sum of the flow absorbed by users (Q_{S1}, Q_{S2}, Q_{S3}), internal pump leakage flow according to the coefficient C_{LkP} and flow Q_{RV} drained towards the tank through the pressure relief valve, acts on pump and lines hydraulic capacity C_P and on accumulator capacity C_A by varying the supply pressure P_S limited to the hydraulic fluid vapour pressure P_V; by the difference between the P_S pressure and the return pressure P_R the differential pressure P_{SR} that the plant delivers to the users is obtained. The last acts on the pressure relief valve modeled as a first order subsystem (the moving element inertia can be considered slightly influent on its dynamic behavior) having τ as time constant, $1/k_{ARV}$ as static gain, x_{RV0} as spring compression in preload condition; the result is the pressure relief valve flapper displacement X_{RV} and, as a consequence, its flow area through the coefficient $C_D \cdot A_{xRV}$.

The last, together with P_{SR}, by means of the flow equation through an orifice, gives the Q_{RV} crossing the pressure relief valve, reported as a feedback on the flow balance regarding the hydraulic capacity. The above-mentioned models have been implemented in a dynamic simulation program capable of analyzing the behavior of the entire system under different conditions and functional configurations [7].

On the basis of the users project requirements and according to defined parametric studies and initial assumptions, possible optimal sizing of the complete system in three different configurations has been attained: irreversible actuators of users 2 e 3; reversible actuators of users 2 e 3 and reversible actuators of users 2 e 3 with ramp generator acting on the command input.

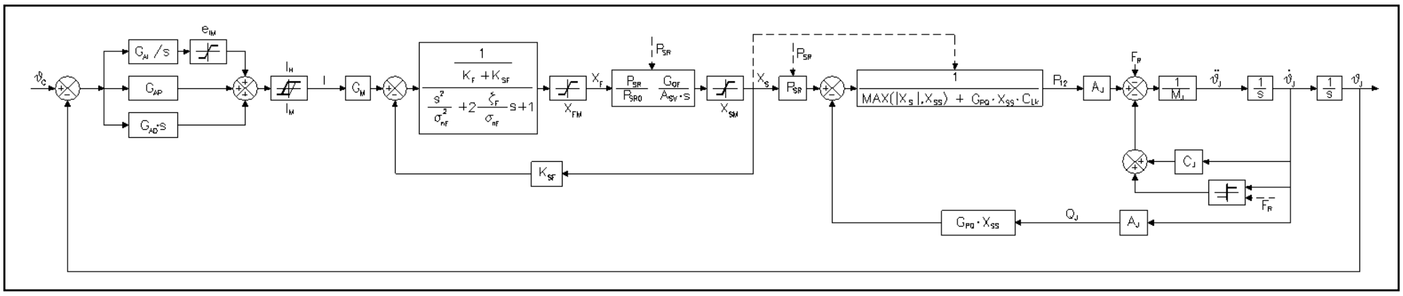

Figure 2. USER 1 block dyagram.

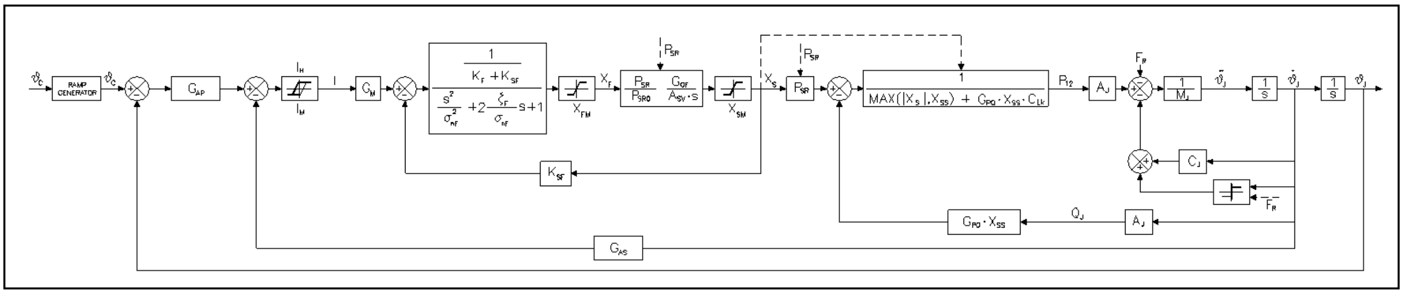

Figure 3. USER 2 and 3 block dyagram.

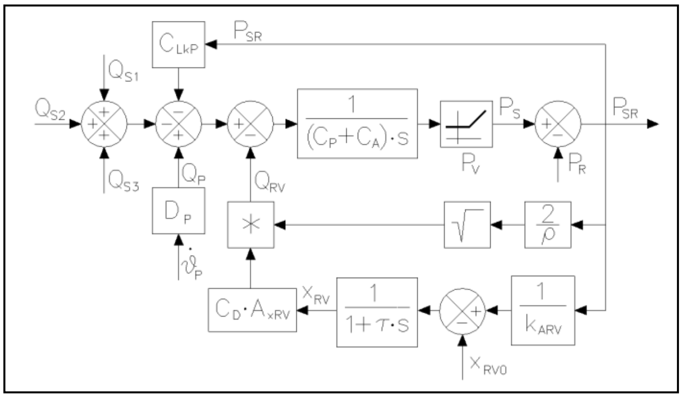

Figure 4. Schematic of the hydraulic system and related users.

V. DIFFERENT CONFIGURATION BEHAVIOUR

Fig. 5 shows the simulation concerning sequenced no load actuations of the three users: at Time = 0 s, when the supply pressure P_S (28.9 MPa) in no flow condition reaches a stabilized value, the users 1 and 2 are respectively submitted to the commands $Com_1 = 0 \rightarrow 0.03$ m and $Com_2 = 0 \rightarrow 0.04$ m, whereas at Time = 0.05 s the user 3 is submitted to the command $Com_3 = 0 \rightarrow 0.035$ m.

When users 1 and 2 start, the supply pressure decreases (producing a partial emptying of the hydraulic accumulator) following the flow absorption and levels on a value of 21.4 MPa, yet adequate for a proper actuation and within the setting range of the pressure relief valve R_V (position X_{RV}). When the third user starts, the further increase of the requested flow rate performs the further decrease of the supply pressure to the value of 13.8 MPa (outside of the range of adjustment of the R_V with complete discharge of the hydraulic accumulator): such a value, in this particular case of null load, yet allow the users actuation, albeit at significantly reduced speed; when the users 1 and 2 travels end, the supply pressure goes up again and the valve R_V recovers its adjustment capability, so permitting a faster final portion of the user 3 travel. Following the complete stop of the three users, the hydraulic plant recovers the initial supply pressure value with R_V valve fully open and hydraulic accumulator fully charged. The sizing of the pump can therefore be considered marginally sufficient to satisfy the requested flow. Fig. 6 shows the case relating to sequenced actuations affected by opposing loads and irreversible actuators characterizing users 2 and 3 (the user 1 actuators efficiency is assumed equal to 1): the sequence of control commands is the same as the case of Fig. 5, while, in the same instants of application of the command, step loads of 10 kN to user 1 and 120 kN to users 2 and 3 are applied.

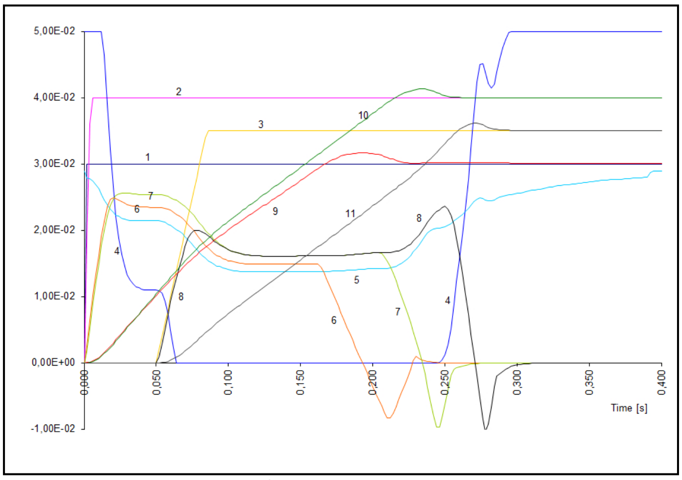

Figure 5. Deferred actuations without loads.

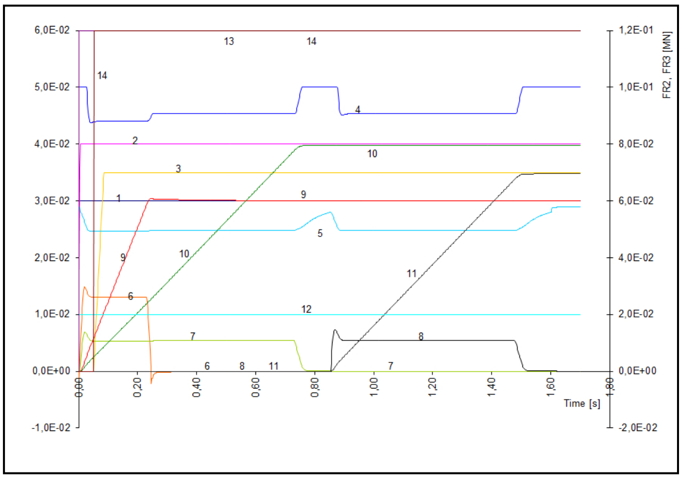

Figure 6. Deferred actuations with opposing loads and irreversible actuators.

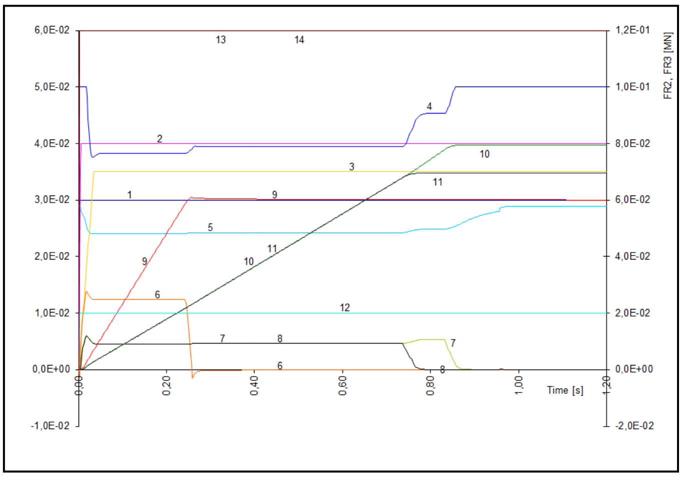

Figure 7. Simultaneous actuations with opposing loads and irreversible actuators.

TABLE I. LEGEND OF THE GRAPHICS

N°	Name	N°	Name
1	Com_1 [m]	8	DX_{J3} [10*m/s]
2	Com_2 [m]	9	X_{J1} [10*m/s]
3	Com_3 [m]	10	X_{J2} [10*m/s]
4	X_{RV} [dm]	11	X_{J3} [10*m/s]
5	P_S [GPa]	12	F_{R1} [MN]
6	DX_{J1} [10*m/s]	13	F_{R2} [MN]
7	DX_{J2} [10*m/s]	14	F_{R3} [MN]

At startup of the first two users the supply pressure decreases due to a smaller (compared to the previous case) flow request stabilizing at a value of 24.7 MPa, still permitting a correct actuation, but not sufficient to the user 3 breakaway. When the user 1 reaches its commanded position and stops (Time = 0.24 s), the supply pressure rises imperceptibly (24.8 MPa) and then more significantly (27.9 MPa) when the user 2 stops (Time = 0.77 s); this pressure rise allows the breakaway of the user 3 (Time = 0.86 s), that can then complete its travel (Time = 1.5 s), after which the initial conditions are restored (28.9 MPa). The presence of opposing loads, limiting the actuation speed, decreases the flow request and consequently also the supply pressure reduction with respect to the condition of stopped users; so the pressure relief valve constantly remains within its useful range of adjustment, excluding the transients characterized by reduced flow request in which the valve is fully open. In this case the pump sizing is therefore insufficient to allow the simultaneous actuation of all users.

Fig. 7 refers to a case quite similar to that of Fig. 6 regardless of the command sequence, which in this case is simultaneous for the three user. Since the start of the users is simultaneous, occurs when the supply pressure assumes its initial value (28.9 MPa), which allows their breakaway; the higher flow rate (compared to the case of Fig. 6) results in a more reduced pressure (24.2 MPa) in steady conditions, slightly increasing their actuation time. In this particular case, the pump sizing appears to be sufficient to perform the simultaneous actuation of all users.

Fig. 8 shows the case relating to sequenced actuations with irreversible actuators and aiding loads characterizing users 2 and 3 with the following sequence of the input commands: at Time = 0 s, when the supply pressure P_S (28.9 MPa) in no flow condition reaches a stabilized value, the users 1 and 2 are respectively submitted to the commands $Com_1 = 0.03 \rightarrow 0$ m and $Com_2 = 0.04 \rightarrow 0$ m, whereas at Time = 0.10 s the user 3 is submitted to the command $Com_3 = 0.035 \rightarrow 0$ m. In the same instants of command input a step opposing load of 10 kN is applied to the user 1 and a step aiding load of 120 kN is applied to users 2 and 3. As a consequence of the actuator irreversibility, the presence of aiding loads produces however a small net opposing load with marginal difference in the users 2 and 3 actuation time, compared to the case of Fig. 5. On the other hand, however, the high opposing load on the user 1 determines a marked slowdown, especially when both users 2 and 3 are travelling.

The pump sizing can therefore be considered marginally sufficient to allow the described actuation. It can be pointed out that the position error of the user 2, when its actuation travel is ended and the user 3 is still travelling, persists in an higher value than this characterizing the condition in which both user 2 and 3 are stopped, as a consequence of the temporarily reduced supply pressure value.

Fig. 9 shows the case relating to sequenced actuations with reversible actuators and opposing loads characterizing users 2 and 3: the sequence of the input commands is the same as the case of Fig. 6. The reversibility of the actuators results in higher efficiencies that allow the user 3 breakaway at the command application time, as well as higher actuation rates. The pump sizing can therefore be considered acceptably sufficient to allow the simultaneous actuation of the three users.

Fig. 10 shows the case relating to sequenced actuations with reversible actuators and aiding loads characterizing users 2 and 3: the sequence of the input commands is the same as in Fig. 8. The reversibility of the actuators results, in presence of aiding loads, in a net aiding load able to increase somewhat the actuation speed (compared to the case of Fig. 8); these interactions between the users markedly impair the proper operation of the user 1, characterized by an opposing load and therefore forced to go back under load when the supply pressure drops below a defined value.

More in detail, when the pressure drops below the value at which the hydraulic accumulator is totally discharged and pressure relief valve is completely closed, the reduced hydraulic capacity of the pipes has an instabilizing effect on the system, which can produce an oscillatory behavior regarding both pressure supply (with possible temporary cavitation affecting the supply pipe itself) and users speeds: in fact, the aiding loads, as a consequence of the low friction level acting on the users 2 and 3 motion transmissions, produce high actuation speed combined with required flow rate significantly higher than that delivered by the pump, resulting in a marked supply pressure drop.

The consequence is a rapid users speed and flow reduction; the flow drops below that provided by the pump and consequently the pressure grows: this produces a new speed recovery and so on in an oscillatory phenomenon slightly damped. The pump sizing can therefore be considered absolutely insufficient to allow the simultaneous actuation of the three users; even a pump cavitation phenomenon may occur and the user 1 is no longer able to perform its task.

Fig. 11 reports a case similar to Fig. 10, with the only difference consisting of the presence, in the control logic of the users 2 and 3, of a command ramp generator limiting the maximum command time rate. It involves the intrinsic capability of limiting the maximum actuation rate performed even in the most unfavorable aiding load conditions; this results in a reduced flow request that maintains a higher value of the supply pressure (compared to the case of Fig. 10) allowing the proper actuation of user 1. The pump sizing can therefore be considered clearly sufficient to allow the simultaneous actuation of the three users.

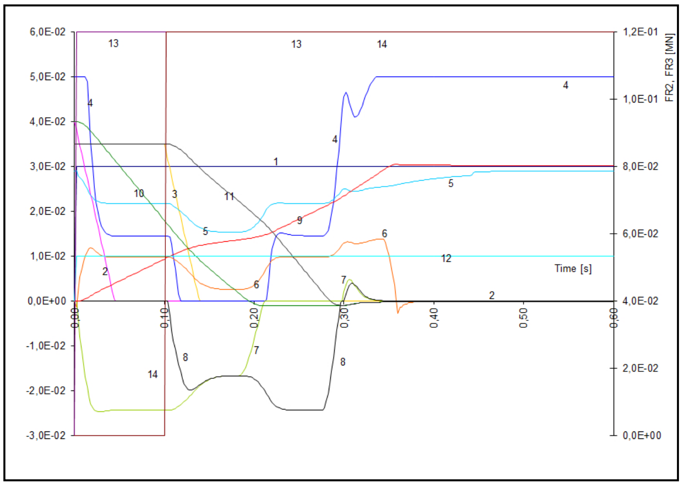

Figure 8. Deferred actuations with aiding loads and irreversible actuators.

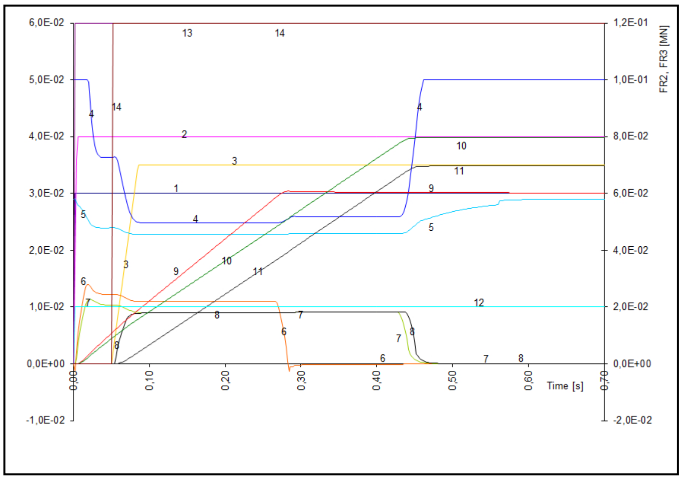

Figure 9. Deferred actuations with opposing loads and reversible actuators.

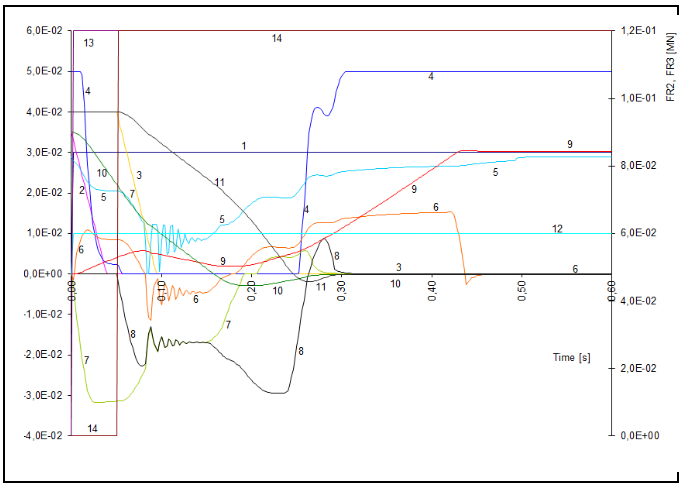

Figure 10. Deferred actuations with aiding loads and reversible actuators.

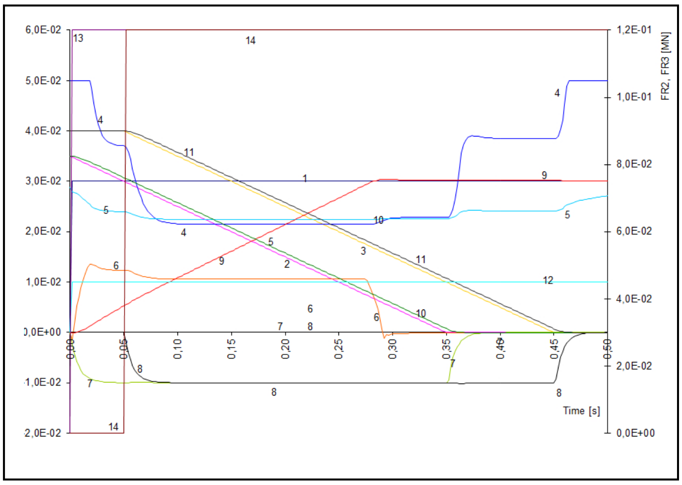

Figure 11. Deferred actuations with aiding loads, reversible actuators and command ramp generator.

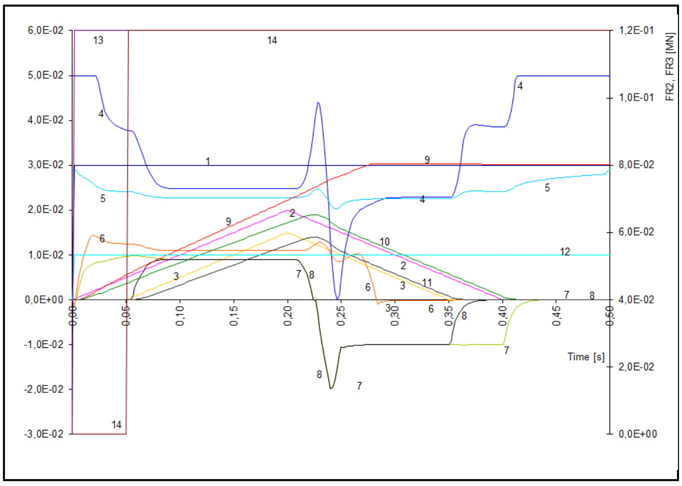

Figure 12. Deferred actuations with aiding / opposing loads, reversible actuators and command ramp generator.

Fig. 12 repeats the case of Fig. 11, with the only difference consisting of the inversion (return to zero), at Time = 0.2 s, of the users 2 and 3 input command retaining the previous load values, so turning from opposing to aiding. The proper actuation of all three users allows to evaluate the benefits performed by the command ramp generator, which prevents any type of malfunctions regarding the entire hydraulic system both in aiding and opposing conditions, as well as in the event of a sudden command reversion. The pump sizing can therefore be considered clearly sufficient to allow the simultaneous actuation of the three users.

VI. CONCLUSION

By the survey of the above reported simulations, referred to the analysis of the dynamic behavior of different system configurations in similar operative conditions, it is deduced as apparently minor design options regarding the users architecture (such as the employment of reversible or irreversible actuators, the possible use of command ramp generators) may play a significant role in the behavior of the entire system, affecting on its sizing, especially with regard to the pump. The above considerations can help the designer in the hydraulic system architecture selection with particular regard to the prevention of improper users interactions.

VII. NOMENCLATURTE

A_J Jack working area
A_{SV} Servovalve second stage driving area
A_{xRV} Ratio between passageway area and flapper displacement of the pressure relief valve
C_A Hydraulic accumulator capacity
C_D Generic discharge coefficient
C_{Lk} Leakage coefficient of the servovalve-actuator assembly
C_{LkP} Pump leakage coefficient
$Com_{1,2,3} = \theta_C$ User 1,2,3 input command
C_P Pipes hydraulic capacity
C_J Jack damping coefficient
D_P Pump displacement [m^3/rad]
$DX_{J1,2,3} = d\theta_J/dt$ User 1,2,3 actuation rate
e_{IM} Integrator output maximum value
$F_{R1,2,3} = F_R$ User 1,2,3 external load
G_{AD} Position derivative gain
G_{AI} Position integrative gain
G_{AP} Position proportional gain
G_{AS} Speed loop gain
G_M Servovalve torquer gain
G_{PQ} Servovalve second stage pressure to flow gain ratio
G_{QF} Servovalve first stage flow gain in reference conditions ($P_{SR} = P_{SR0}$)
I Servovalve input current
I_M Servovalve current in saturation conditions
k_{ARV} Ratio between pressure relief valve spring stiffness and shutter area
K_F Servovalve first stage elastic hinge stiffness
K_{SF} Servovalve internal feedback spring stiffness
M_J Mass of the actuator-user assembly
P_R Hydraulic system return pressure
P_S Supply pressure
P_{SR} Differential supply to return pressure
P_{SR0} Reference value of P_{SR}
P_V Hydraulic oil vapor pressure
P_{12} Differential pressure acting on the actuator
Q_P Pump flow
Q_{RV} Pressure relief valve flow
$Q_{S1,2,3}$ Users 1,2,3 requested flows (respectively)
X_F Servovalve first stage position
X_{FM} Servovalve first stage end of travel
X_{RV} Pressure relief valve shutter position
X_{RV0} Ratio between pressure relief valve spring preload to stiffness
X_S Servovalve second stage position
X_{SM} Servovalve second stage end of travel
X_{SS} X_s value at which P_{12} reaches saturation condition
$X_{J1,2,3}$ User 1,2,3 actuator position
σ_{nF} Servovalve first stage undamped circular frequency
ζ_F Servovalve first stage damping ratio
ρ Oil density
τ Pressure relief valve time constant

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