HYDRAULIC FLUID VALIDATION THROUGH MODELING AND SIMULATION OF A SERVO ACTUATOR SYSTEM

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Abstract. *The hydro-mechanical servo actuator system response can be degraded by the hydraulic fluid, when used as power transmission medium. The study consisted on modeling the servo actuator system, which considers non-linear characteristics such as servo valve variable flow, actuator friction. Furthermore, the dynamic response is analyzed when applied phosphate esther-based fluids of different properties. Three hydraulic fluids 'Skydrol LD-4',' Skydrol 500B-4' and 'Skydrol 5' were validated for the servo actuator at a determined temperature. The simulations show that the hydraulic fluids Skydrol LD-4 and Skydrol 5 allow faster system responses when compared to the Skydrol 500B-4. This is explained by the fluid density, when it is lower, the fluid peak pressure is higher, which decreases the time to the piston achieve its commanded position.*
Keywords*: servo actuator, servo valve, hydraulic fluid.*

1. INTRODUCTION

Nowadays, to survive among a competitive market is necessary to innovate in a shorter period, increasing the quality and test security, and reducing costs.

The aerospace industry is an example of competitive market, which requires a continuous product development improvement. The aircraft manufacturers have been exploring the systems models simulation, glimpsing a reduction of physical tests. It allows the manufacturers to develop theirs products in an economical way and in a reduced period of time, using the expensive tests just for products performance validation.

The modeling method has been applied to several aircraft systems, including flight controls system. There are some important aspects that shall be taken into account as real hydro-mechanical system effects e.g. orifice flow for laminar and turbulent conditions [Borutzky, Barnard and Thoma, 2002], pressure fluctuation in hydraulic pipes [Higo, Yamamoto, Tanaka, Sakurai and Nakada, 2000], variable orifice flow [Viall, Zhang, 2000], system performance under different types of fire resistant fluids [Dasgupta, Chattapadhyay, Mondal, 2005], non-linearities related to servo actuator [Joshi, 2005]. Some of these effects have already been analyzed in laboratory tests, contributing to modeling community.

There are some studies to be conducted e.g. evaluate the system performance under hydraulic fluid temperature variation, considering all the servo actuator non-linearities, comparative study on how the hydraulic fluids affect system dynamic response.

The system designer shall have a complete knowledge of the selected hydraulic fluid properties as well as its effects, in respect to the compatibility between the hydraulic components and fluid as o-rings and the compliance of the system dynamic behavior and considering the system non-linearities.

The study consists on investigating the hydro-mechanical actuator system response when submitted to hydraulic fluids with different properties, considering non-linear aspects related to actuator and variable orifice flow for laminar and turbulent conditions.

2. PROBLEM FORMULATION

Figure 1 presents the schematic of a flight control system described in Gritti, 2004, considering a hydro-mechanical servo actuator.

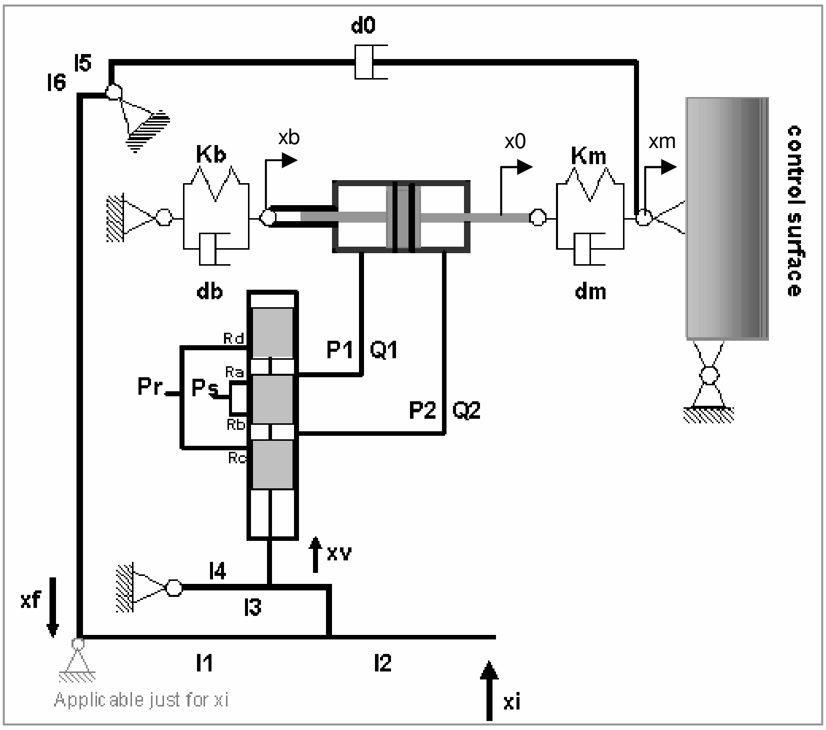

Figure 1 Schematic of a flight control system.

From Figure 1, it is observed that the system is composed by mechanical and hydraulic components as levers mechanism, hydraulic servo valve, hydraulic actuator and a control surface.

The command action reaches the input lever (xi), which is transmitted through the levers mechanism to the valve spool, displacing it to up or down (xv). This transmission ratio is given by Eq. (1).

$$xv = \frac{l1}{l1 + l2} * \frac{l4}{l3} * xi \tag{1}$$

where:

l1, l2, l3, l4: titanium levers [m]
xi: input displacement [m]
xv: valve spool displacement [m]

The valve spool displacement generates the valve orifices restrictions opening, which is composed by a 4-port way. One port is fed by hydraulic fluid under a determined pressure (Ps) generated by the Hydraulic Power Generation System, the other one is way to give away the hydraulic fluid (Pr) that is not used, and the others are connected to the actuator ports (P1, P2). When the valve spool is centralized, there is no flow and the modulated resistances are infinite due to Ra, Rb, Rc and Rd are function of xv. When it moves up, (xv) is positive and the modulated resistances Ra and Rc become finite, allowing the fluid flow pass through the orifices as expressed by Eq. (2) (Qa) and Eq. (4) (Qc). On the other hand, the modulated resistances Rb and Rd become infinite, there is no fluid flow. The inverse can be applied to the valve when its spool moves down, (xv) is negative and the modulated resistances Rb and Rd become finites, allowing the fluid flow pass through the orifices as expressed by Eq. (3) (Qb) and Eq. (5) (Qd). On the other hand, the modulated resistances Ra and Rc become infinites, there is no fluid flow. The flow equations below were modeled based on Borutzky, Barnard and Thoma, 2002, which considers orifice flow for laminar and turbulent conditions. The fluid viscosity and density are properties that affect the fluid flow and depend on the temperature and pressure.

The clearances leakage in the servo valve design is acceptable and represented by $Gleak.\Delta P / \nu$ but it shall be limited by the design requirement in order to maintain the system performance [Bizarria, 2009]. It helps to avoid the friction between movable structures as in this case, the valve spool and the valve itself.

$$Qa = \left(cturb * w * xv * \sqrt{\frac{2}{\rho} * |Ps - P1| + \left(\frac{\nu * Rt}{4 * cturb * w}\right)^2} - xv * \nu * \frac{Rt}{4} \right) * sign(Ps - P1) + Gleak \frac{.(Ps - P1)}{\nu} \quad (2)$$

for xv > 0

$$Qb = \left(cturb * w * xv * \sqrt{\frac{2}{\rho} * |P1 - \mathrm{Pr}| + \left(\frac{\nu * Rt}{4 * cturb * w}\right)^2} - xv * \nu * \frac{Rt}{4} \right) * sign(P1 - \mathrm{Pr}) + Gleak. \frac{(P1 - \mathrm{Pr})}{\nu} \quad (3)$$

for xv < 0

$$Qc = \left(cturb * w * xv * \sqrt{\frac{2}{\rho} * |P2 - \mathrm{Pr}| + \left(\frac{\nu * Rt}{4 * cturb * w}\right)^2} - xv * \nu * \frac{Rt}{4} \right) * sign(P2 - \mathrm{Pr}) + Gleak. \frac{(P2 - \mathrm{Pr})}{\nu} \quad (4)$$

for xv > 0

$$Qd = \left(cturb * w * xv * \sqrt{\frac{2}{\rho} * |Ps - P2| + \left(\frac{\nu * Rt}{4 * cturb * w}\right)^2} - xv * \nu * \frac{Rt}{4} \right) * sign(Ps - P2) + Gleak. \frac{(Ps - P2)}{\nu} \quad (5)$$

for xv < 0

where:

Qa, Qb, Qc, Qd: flow through orifice [m^3/s]
cturb: =0.61 discharge coefficient applied to high Reynolds values
w: area gradient [m^2/m]
ρ: fluid density [kg/m^3]
ν: kinematics viscosity [m^2/s]
Rt:=9.33 [Borutzky, Barnard, Thoma, 2002, p. 146]
Ps: forward pressure [Pa]
Pr: return pressure [Pa]
P1, P2: valve pressure [Pa]
Gleak: laminar leakage conductance in the valve clearances

The piston movement model is powered by the pressure difference between actuator chambers over the piston area as presented in Eq. (6). In the same way, it is reduced by non-linear forces as friction between the cylinder and piston [Kuster, 1996], and collision force between the actuator cylinder strokes and piston area given by Eq. (7), Eq. (8) and Eq. (9). The reaction forces given by mechanic connection between the piston lever and surface rod kinematics reflect in the piston movement, reducing the piston force.

$$Mp * \ddot{x}0 = Ap * PL - Km * (x0 - xm) - dm * (\dot{x}0 - \dot{x}m) - Ffriction + Fcol \quad (6)$$

$$Ffriction = Fc * \tanh(slope * (\dot{x}0 - \dot{x}b)) + dv * (\dot{x}0 - \dot{x}b) \quad (7)$$

$$Fcol = -Kc * (x0 - xb) - dc * \min([(\dot{x}0 - \dot{x}b), 0]) \quad (8)$$

for $x0 - xb < stroke$, stroke< 0

$$Fcol = -Kc * ((x0 - xb) - stroke) - dc * \max([(\dot{x}0 - \dot{x}b), 0]) \quad (9)$$

for $x0 - xb > stroke$, stroke > 0

$$PL = P1 - P2 \quad (10)$$

where:

Mp: piston mass [kg]
Ap: piston area [m2]
PL: pressure difference between actuator chambers [Pa]

Ffriction: friction forces between cylinder and piston [N]
Fc: static friction coefficient [N/m]
Slope: friction Coulomb slope
dv: viscous friction coefficient [N.s/m]
Fcol: collision force [N]
Kc: actuator piston and cylinder stiffness [N/m]
dc: actuator piston and cylinder damping [N.s/m]
stroke: actuator cylinder length [m]
x0: piston displacement [m]
xb: actuator cylinder displacement [m]
$\dot{x}0$: piston speed [m/s]
$\dot{x}b$: actuator cylinder speed [m/s]
Km: rigidez da conexão atuador superfície [N/m]
db: amortecimento do atuador estrutura [N.s/m]
dm: amortecimento do atuador superfície [N.s/m]
$\ddot{x}0$: piston acceleration [m^2/s]

The flow through actuator is given by Eq. (11). It considers the actuator internal and external leakages, non-linearities related to fluid compressibility.

$$QL = Ap * (\dot{x}0 - \dot{x}b) + \frac{V}{\beta} * \dot{P}L + Cli * PL \tag{11}$$

where:

Ap: piston area [m^2]
Cli: actuator internal and external leakage coefficient [$m^3/s/Pa$]
β: bulk modulus [Pa]
QL: flow through the actuator [m3/s]
V: hydraulic fluid volume [m3]
$\dot{P}L := \frac{dPL}{dt}$ load pressure varying in time [Pa/s]

The actuator cylinder model is represented by Eq. (12) and is powered by its mechanical connection with the aircraft structure as structure stiffness and connection damping. The equation also considers forces reaction generated by piston movement and its non-linearities.

$$Mb * \ddot{x}b = -Ap * PL + Kb * xb + db * \dot{x}b + Ffriction - Fcol \tag{12}$$

where:

Mb: actuator cylinder mass [kg]
Kb: actuator structure connection stiffness [N/m]
db: actuator structure connection damping [N.s/m]
xb: actuator cylinder displacement [m]
$\dot{x}b$: actuator cylinder speed [m/s]
$\ddot{x}b$: actuator acceleration [m^2/s]

For the surface model, it has not been considered the kinematics between the piston rod and the surface. It was simplified in order to reduce it to a mass load connected to piston rod. Equation (13) considers the mass dynamic related to the structure connection as stiffness and damping.

$$M * \ddot{x}m = Km * (x0 - xm) + dm * (\dot{x}0 - \dot{x}m) - d0 * \dot{x}m \tag{13}$$

where:

M: load mass [kg]
$\ddot{x}m$: surface acceleration [m^2/s]

The surface movement displaces the feedback levers mechanism, which will be summed to the input command, modifying the valve spool position, described by the Eq. (14).

$$xv = \frac{l2}{l1+l2} * \frac{l4}{l3} * \frac{l6}{l5} * xm \tag{14}$$

where:

l5, l6: titanium levers [m]

Equation (1) and Eq. (15) are simplified to:

$$xv = ki.xi - kf.xm \tag{15}$$

where:

$$ki = \frac{l1}{l1+l2} * \frac{l4}{l3} \tag{16}$$

$$kf = \frac{l2}{l1+l2} * \frac{l4}{l3} * \frac{l6}{l5} \tag{17}$$

2.1 CASES SIMULATONS

The studies are based on the hydro-mechanical system's characteristics, whose parameters are presented in Appendix A Table A.1, evaluating the hydraulic fluid properties that influence the rod displacement. It has been generated three systems models, by changing the hydraulic fluid properties, as bulk modulus, density and kinematics viscosity as presented in Appendix A Table A.2.

The system model is simulated in closed loop, considering a step as a reference equivalent to a rudder pedal deflection of 9 degrees. This signal is summed to the feedback signal resulting in the valve spool displacement as shown in Fig. 2, that controls the valve restriction, by opening or closing it. In this case study the valve opens suddenly and closes as soon as the piston actuator reaches the cylinder stroke, which is connected to a rod that reflects the piston final position as well as its speed, presented by Fig. 3, Fig.4 and Fig. 5, Fig. 6, respectively. This process controls the orifice opening allowing the fluid flows through the valve and the cylinder, generating pressure difference between the cylinder actuator chambers as shown in Fig. 7 and Fig. 8.

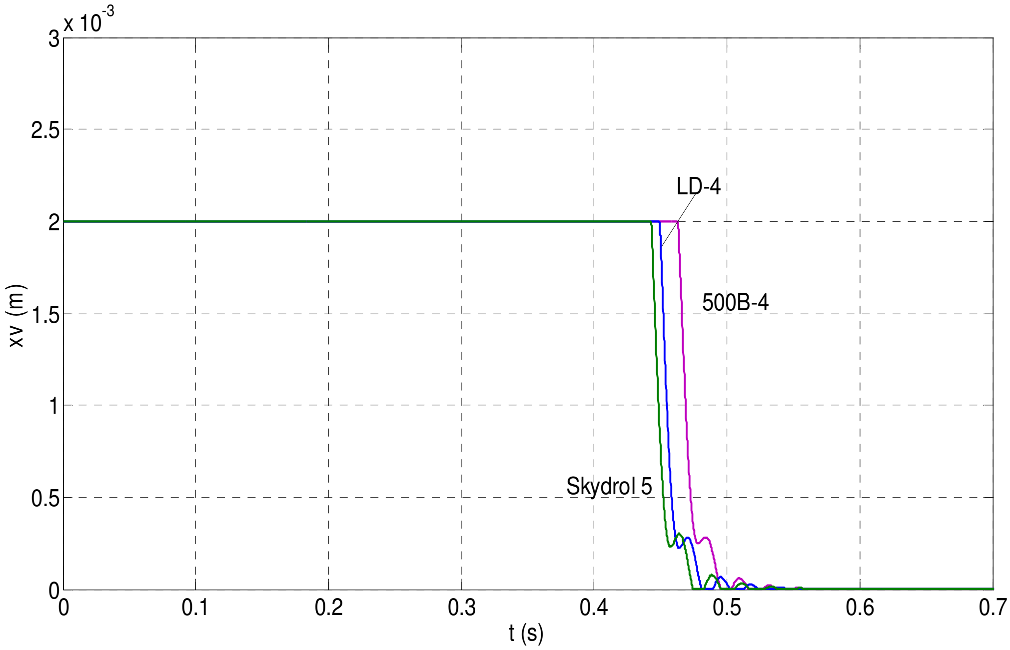

Figure 2 Spool valve displacement comparison.

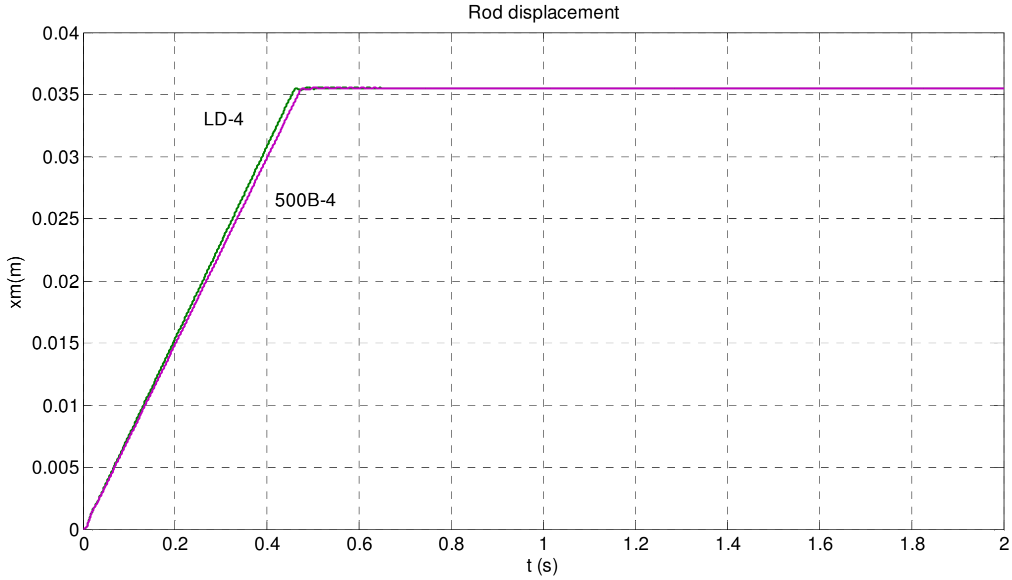

Figure 3 Rod displacement comparison between Skydrol LD-4 and Skydrol 500B-4 hydraulic fluid.

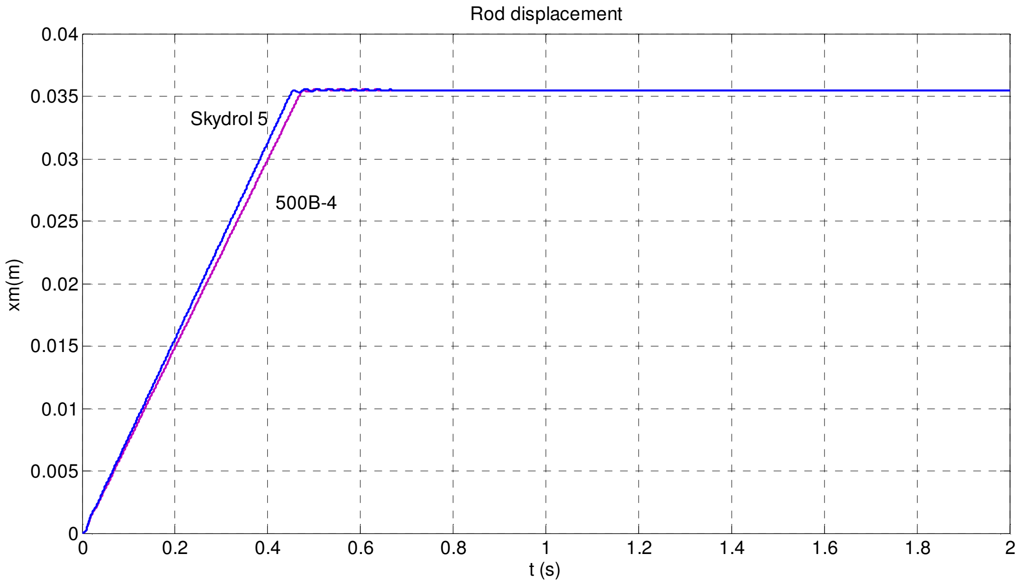

Figure 4 Rod displacement comparison between Skydrol 5 and Skydrol 500B-4 hydraulic fluid.

The Fig. 5 and Fig. 6 present the comparison of the dynamic response of the rod speed using Skydrol 5 and Skydrol LD-4 hydraulic fluid with respect to another hydraulic fluid Skydrol 500B-4, a hydraulic fluid commonly used in the aircraft fleet. As soon as the valve closes, the rod speed goes to zero. After valve closing, the rod speed oscillates due to the low resistive load at the actuator end and the low leakage flow of the actuator. It is also seen that the rod speed peak is higher when using the Skydrol 5 and Skydrol LD-4, reaching the zero earlier than the model using Skydrol 500B-4. Fig. 2.2 and Fig. 2.3 show the speed impact in system response time, in both figures the Skydrol 5 and Skydrol LD-4 present a better system performance.

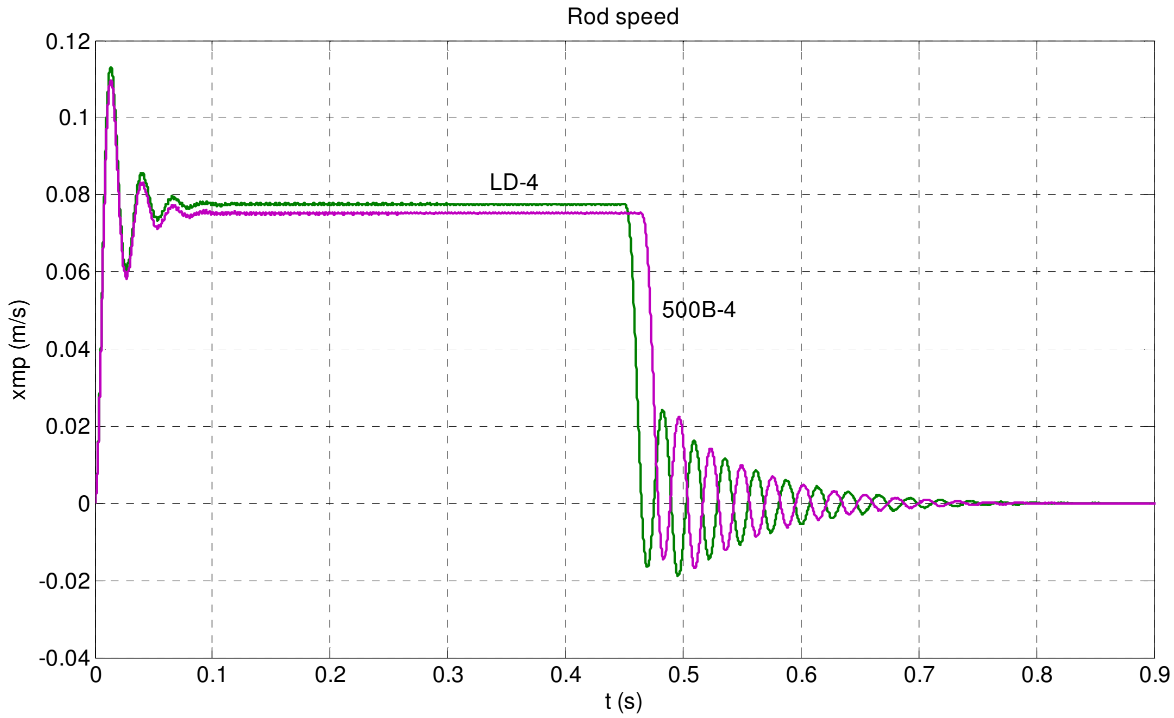

Figure 5 Rod speed comparison between Skydrol LD-4 and Skydrol 500B-4 hydraulic fluid.

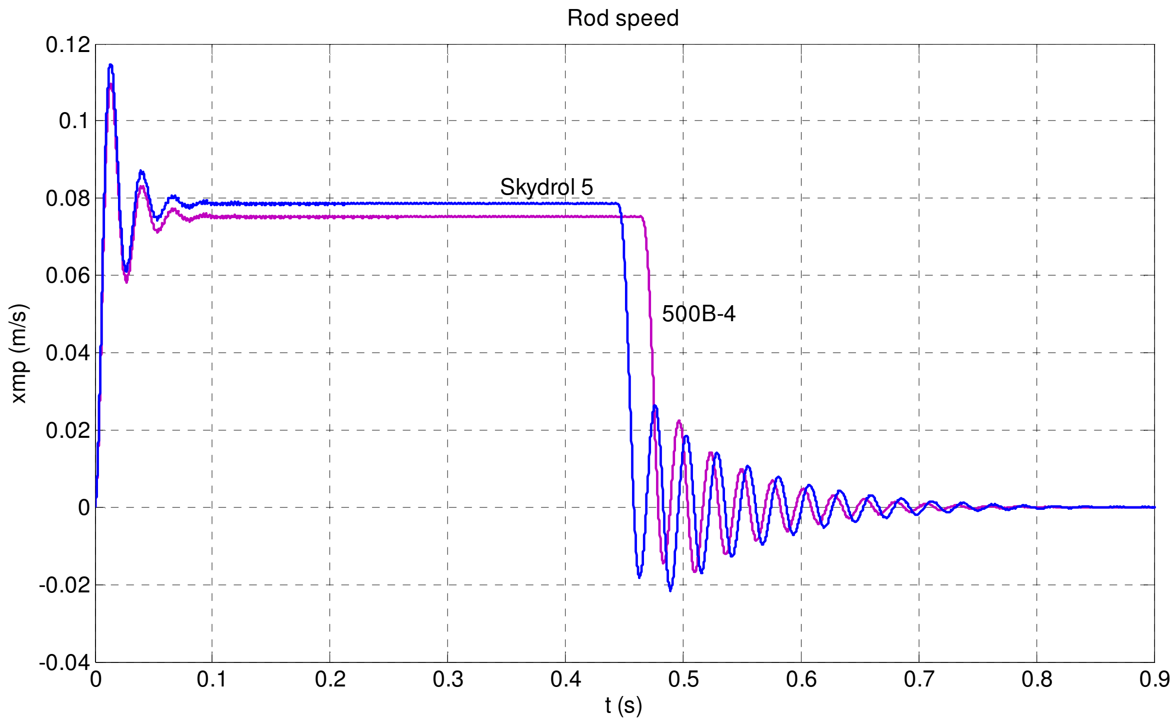

Figure 6 Rod speed comparison between Skydrol 5 and Skydrol 500B-4 hydraulic fluid.

Figure 7 and Fig. 8 show pressure difference between actuator chambers. The high peak pressure due to the effective bulk stiffness of the fluid at the actuator chamber and the load inertia attached to rod end. When the valve is closed, there is a oscillatory behavior generated by the kinetics and potential energies changed among piston and hydraulic fluid, to continue oscillating until leakage loss dissipate the energy involved [Merrit, 1967].

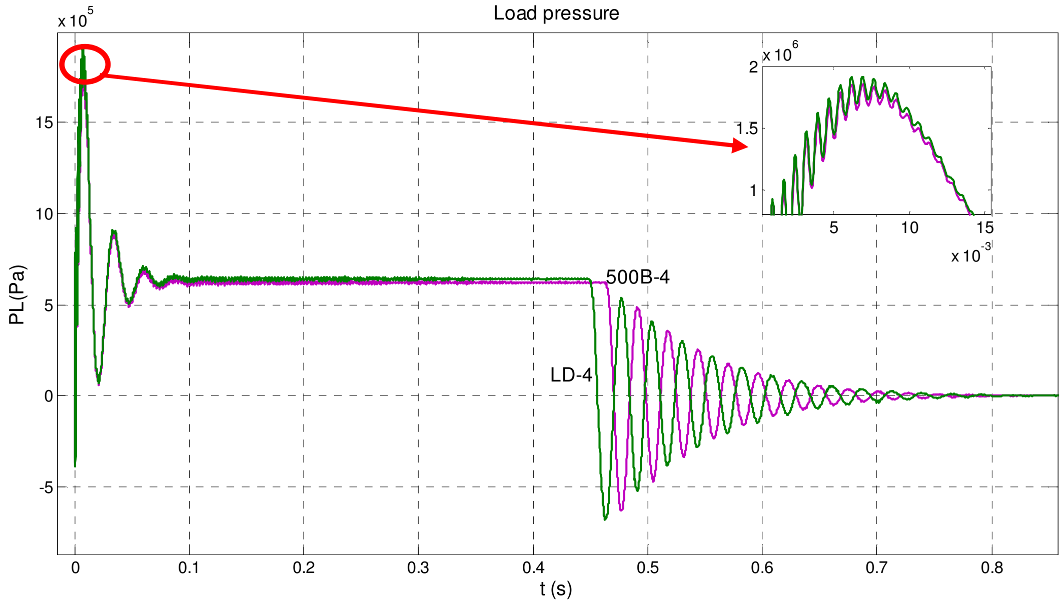

Figure 7 Load pressure difference comparison between Skydrol LD-4 and Skydrol 500B-4 hydraulic fluid.

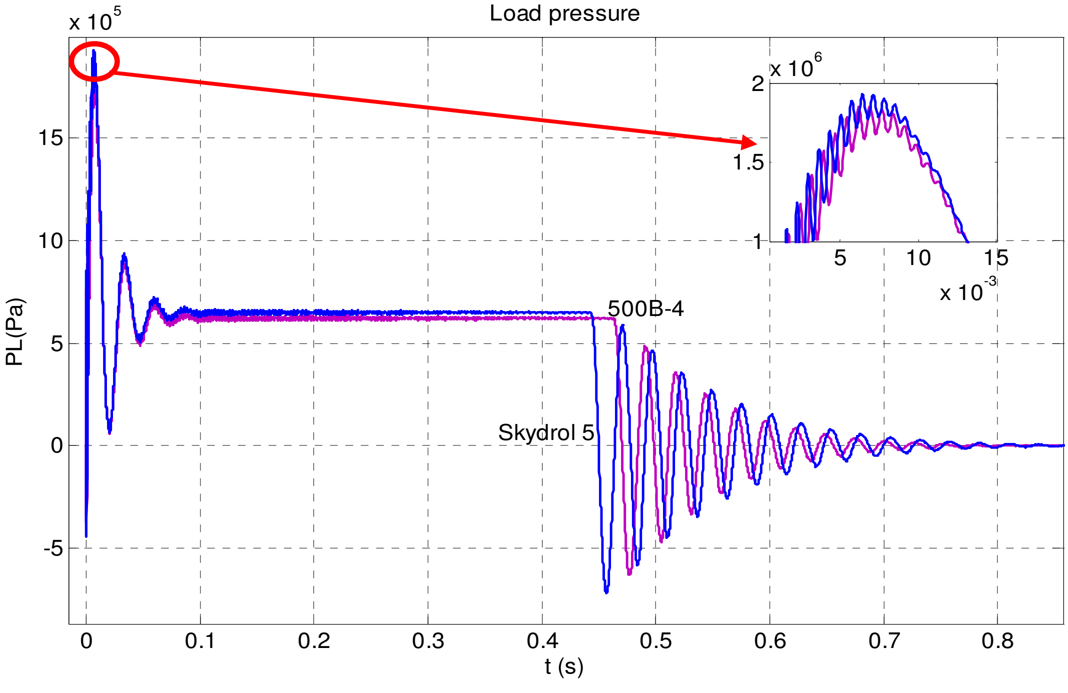

Figure 8 Load pressure difference comparison between Skydrol 5 and Skydrol 500B-4 hydraulic fluid.

The simulation results indicate that when the density of the fluid decreases, the peak pressure and the fluctuation of the fluid pressure increases. It decreases the system time to reach zero in the pressure difference and rod speed, and it also decreases the system time to reach the stroke cylinder actuator.

This system behavior difference is due to the fluids properties, since the other parameters are all the same for the three models.

3. CONCLUSION

The present study evaluates the dynamic system response for a hydro-mechanical servo actuator system. The study presents the case study modeling as well as mathematical formulation considering the real system effects such as fluid compressibility, viscous and friction damping and servo valves nonlinearities. As simulation result, it was shown for rod position, actuator rate, and pressure difference between actuator. The simulation results show that the hydraulic fluid properties affect the dynamic system. Although the fluid of high density damps the system transient response, it delays the steady state response.

For future studies it is recommended to consider a change of physical properties of the hydraulic fluid due to trapped air evaluating the bulk modulus and system performance, improvements applied to the actuator model.

APPENDIX A

Table A.1 Data used for simulation

Parameter	Value	Unit	Description
ki	0.2591	-	Coefficient for input signal
kf	2.496	-	Feedback coefficient
Mb	1.0	kg	Actuator cylinder mass
Kb	3.40×10^{6}	N/m	Actuator structure stiffness
db	0.00	N.s/m	Actuator structure damping
Kc	1.0×10^{7}	N/m	Stiffness during colision with cylinder stroke
dc	1.0×10^{4}	N.s/m	Damping during colision with cylinder stroke
stroke	35.6×10^{-3}	m	Cylinder stroke
Fc	123	N/m	static friction coefficient
slope	1000	-	Coulomb friction slope
dv	3.40×10^{8}	N.s/m	Slip friction
Mp	3.5	kg	Piston mass
Km	1.98×10^{7}	N/m	Actuator surface stiffness
dm	0.0	N.s/m	Actuator surface damping
d0	1043.26	N.s/m	Structure damping
M	45.00	kg	Surface mass
A1_pistão, A2_pistão	4.9×10^{-4}	m^2	Piston area
Ps	20.7×10^{6}	Pa	supplied pressure
Pr	0.00	Pa	Return pressure
Cli1, Cli2	5.0×10^{-17}	$m^3/s/Pa$	Actuator leakage coefficient
Rt	9.33	-	Transition Reynolds
cturb	0.61	-	Discharge coefficient for high Reynolds values
Cd	0.61	-	Discharge coefficient
xmax	2.0×10^{-3}	m	Maximum valve spool module (xv)
Gleak	1.4×10^{-17}	$m^3/s/Pa$	Valve leakage coefficient

Table A.2 Fluid properties used in the simulation (Skydrol catalogue)

Fluid type	Density (kg/m3)	Kinematics viscosity (m2/s)	Bulk modulus (Pa)
Skydrol 500B-4	1061.68	1.15E-05	1.89E+09
Skydrol LD-4	1000.57	1.14E-05	1.88E+09
Skydrol 5	974.20	9.23E-06	2.20E+09

4. REFERENCES

1. Bizarria, C. O., 2009. "Prognóstico de Falhas no Atuador do Leme da Aeronave Embraer-190", Masters Thesis – Instituto Tecnológico de Aeronáutica, São José dos Campos. (Msc. Thesis)
2. Borutzky, W., Barnard, B. and Thoma, J., 2002. "An Orifice Flow Model for Laminar and Turbulent Conditions", Simulation Modeling Practice and Theory, p.141-152.
3. Dasgupta, K.; Chattapadhyay, A.; Mondal, S. K., 2005 "Selection of Fire Resistant Hydraulic Fluids Through System Modeling and Simulation". Simulation Modelling Practice and Theory, p.1-20.

4. Green, W. L., 1985. "Aircraft Hydraulic Systems: An Introduction to the Analysis of Systems and Components". Great Britain: John Wiley & Sons Ltd, 137p.
5. Gritti, M., 2004. "Especificação da Arquitetura e Análise de Desempenho de um Sistema de Leme", Masters Thesis – Instituto Tecnológico de Aeronáutica, São José dos Campos.
6. Higo, H., Yamamoto, K., Tanaka, K., Sakurai, Y., Nakada, T., 2000. "Bondgraph Analysis on Pressure Fluctuatio in Hydraulic Pipes**".** Industrial Electronics Society, 2000. IECON 2000. 26th Annual Confjerence of the IEEE**.**
7. Joshi, A., 2005. "Modelling of Flight Control Hydraulic Actuators Considering Real System Effects". Engineering Simulators Uses and Techniques III, AIAA-2005-6297.
8. Kuster, H. E. 1996. "Projeto e Análise de um Servo Atuador Hidromecânico para Aplicação Aeronáutica**",** Masters Thesis – Instituto Tecnológico de Aeronáutica, São José dos Campos.
9. Merrit, H. E., 1967. "Hydraulic Control Systems". New York: John Wiley & Sons Inc.
10. Viall, E. N., Zhang, Q., 2000. "Determining the Discharge Coefficient of a Spool Valve". Proceedings of American Control Conference. Chicago, Illinois.

5. RESPONSIBILITY NOTICE

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