PROPOSALS AND COMPARATIVE EVALUATION OF SYNTHETIC NONLINEAR MODELS IN FLIGHT CONTROLS SIMULATION

L. Borello, G. Villero
Department of Aerospace Engineering - Politecnico di Torino
Corso Duca degli Abruzzi 24 – 10129 TORINO)

Keywords: *flight controls, mathematical modelling*

Abstract

The paper proposes two simplified second order mathematical models concerning generic position servomechanisms and discusses their behaviour compared to a high order model considered as a sufficiently accurate reference for an electrohydraulic servo-system; the comparison is furtherly extended to the first order models usually employed in the dynamic simulation of the aeroplane – autopilot – servocontrol system. The results point out that only the proposed non linear second order model is a satisfying answer to the demand of simple, but acceptably accurate, simulation model of the flight controls.

1 Introduction

The today's flight controls consist of a greater and greater number of more and more complex components in order to satisfy more and more strict performances and safety requirements. It requests the necessary availability, for the designer, of very detailed models of single components or specified subsystems; moreover it is appropriate to employ simple but sufficiently fitting models in order to simulate complete systems, particularly when detailed data of the components are unavailable and the computing devices are characterised by limits concerning computing time or performances.
The simple models are also particularly appropriate in preparing a system monitoring software able to operate in flight conditions (real time), when the airborne computer job is particularly burdensome.

2 Aims of the work

On the basis of the above mentioned considerations, the present paper proposes two simplified mathematical models concerning generic position servomechanisms independent from the power actuation source (electrical or hydraulic).
The proposed models essentially compute the instantaneous rate and position of the actuator – aerodynamic surface assembly as a function of the commanded position and applied load.
By means of dedicated computer programs, this work tests the capability of the proposed models (Figs. 3-4): together with some existing models (Figs. 1-2), they are compared with a further fifth order non-linear and medium complexity reference dynamic simulation model (Fig. 5). The last represents (with a good accuracy) the dynamic of a position electrohydraulic servomechanism essentially composed by control valve, linear hydraulic piston (with leakage, viscous and dry friction) with connected moving surface, proportional position and position-rate loops.
Hence, aims of the work are the description and the validation of the simplified proposed models, by means of the comparison between their dynamic behaviours and the reference model one.

3 Simplified models of flight controls

In terms of dynamic behaviour, the considered (existing or proposed) models represent the position servomechanism of a generic flight control as a system of:

1. linear first order insensitive to the load act

ing on the moving surface (existing model - Fig. 1);
2. non - linear (saturation) first order insensitive to the load acting on the moving surface (existing model - Fig. 2);
3. linear second order sensitive to the load acting on the moving surface (proposed model - Fig. 3);
4. non - linear (saturation) second order sensitive to the load acting on the moving surface (proposed model - Fig. 4).

The first order models (negligible inertia) are characterised by defined values of position gain (positioning stiffness under loaded condition KJ) and time constant (CJ/KJ) or viscous constant CJ, the second order ones (significant inertia) by defined values of position gain KJ, natural frequency or inertia constant MJ, non – dimensional damping or viscous constant CJ.

The non-linearities of the first order model consist of the limitations concerning the maximum actuation rate and the maximum available travel.

The non-linearities of the second order model come from the ability of taking into account the ends of travel of the controlled element and the limitation of the maximum drive force that the hydraulic actuator can develop in static conditions; this latter issue involves the consequent identification of the maximum unloaded actuation rate of the aerodynamic surface and of the maximum drive torque that the jack can apply to such surface in steady state conditions (stall).

The command input variable is formed by the demanded moving surface angle Com, the disturbance input variable (present in the second order models only) by the aerodynamic load FR, whereas the output variable is seen in terms of elevator angle XJ actually obtained, consequently being intended as "position error" the difference between the demanded surface angle and the actual angle Com-XJ. It must be noted that the demanded and actual surface angles are expressed in terms of piston displacement and the aerodynamic load in terms of axial force acting on the above mentioned piston.

The elastic constant KJ represents the loaded positioning stiffness of the servocontroller, due to the presence of the position loop; if its control law is exclusively proportional, this can be modelled with proper accuracy by imposing a defined value to the ratio between the drive torque generated by the jack onto the moving surface in steady conditions and the (angular) position error that produces it; in fact, the elastic constant represents the above mentioned ratio. Instead, if the position loop is characterised by a control law more sophisticated than the purely proportional law (e.g. proportional with conditioning filters, PID, etc.), some aspects of the servocontroller behaviour, generally of second approximation, cannot be modelled simply through KJ. The elastic constant is chosen in view of obtaining the typical values of loaded positioning stiffness characterising primary flight controls. In particular it is obtained by the block diagram of Fig. 5 as the relationship between the position error and the consequent force produced on the piston.

The viscous constant CJ approximately represents the damping produced on the servocontroller dynamics both by possible control actions (speed loop, if present, subjected to the limitation of the maximum drive force) and by fluid-dynamic actions (consisting of two terms: friction on the mechanical elements assumed as a viscous damping without any force limitation; oil leakage through the control passageways of the valve with an effect which can be only partially assumed as a limitation of the maximum drive force). The viscous constant is chosen in order to obtain appropriate non-dimensional damping values in the second order models; the same value is adopted in the first order models.

In the second order non linear model two different speed loops are considered: the former performs a viscous damping without any force limitation (selected in order to obtain the same no load actuation speed as in the reference model), the latter a viscous damping subjected to the drive force limitation (selected in order to obtain appropriate non-dimensional damping values), as represented in Fig. 4.

The inertia constant MJ of the second order models represents the moment of inertia, reduced to the moving surface shaft, of the group consisting of hydraulic piston, aerodynamic surface and mutual interconnection items, sub jected to the aerodynamic load. The inertia constant is selected in view of obtaining correct values of the servomechanism first natural frequency.

4 Validation of the simplified proposed models

According to the above mentioned mathematical models, a dynamic simulation program has been prepared in order to evaluate the behaviour of the proposed models in different command and load conditions.
Figures 6-7-8-9-10-11-12-13 show the trend of the input command and of the displacements of the hydraulic piston as computed by the different models; in these figures XJ1L, XJ1S, XJ2L, XJ2S and XJ represent respectively the displacements of:

- the linear first order model insensitive to the load of Fig. 1
- the non - linear (saturation) first order model insensitive to the load of Fig. 2
- the linear second order model sensitive to the load of Fig. 3
- the non - linear (saturation) second order model sensitive to the load of Fig. 4
- the fifth order non-linear and medium complexity reference model of Fig. 5.

Figure 6 refers to an input step command from 0 to 0.0001 m with low flow gain and no load. The low flow gain gives a good stability margin to the system, so no overshoot is present in the response; the only reference model, having the ability to compute the effect of the dry friction, shows a consequent steady state position error at the end of the actuation travel. No difference is shown between the displacements of the linear models and the corresponding non linear ones, because the small actuation travel does not involve any saturation. Obviously, at the beginning of the actuation travel, the first order models start move quickly, followed by the second order ones, later followed by the reference model (assumed as the actual behaviour of the hydraulic system). Generally both the existing and especially the proposed models are able to represent with a good accuracy the actual system behaviour.
Figure 7 refers to an input step command from 0 to 0.01 m with low flow gain and no load.
The reference model (XJ) shows a small overshoot which is not present in the responses of the second order models, while this is an intrinsic characteristic for the first order models. The reference model overshoot, absent in the case of a small step command of Fig. 6, is probably caused by the end of travel of the servovalve first stage, which is only engaged in case of large commands. The responses of the linear models are too fast because they are not able to take into account the maximum value of the actuation speed of the system. The first order non linear model XJ1S shows a response in advance with respect to the corresponding response of the reference one, while the non linear second order model XJ2S sufficiently fits XJ. Generally the non linear models are able to represent with an acceptable accuracy the actual system behaviour, while the linear ones are markedly inadequate.
Figure 8 refers to an input step load from 0 to 10 kN with no command and with low flow gain. The first order models, owing to their insensitivity to the load, are not able to represent the piston displacement under the load effect; the second order models show a good compliance with the actual system behavior. The different final position between the second order models and the reference one depends on the inability of the second order models to take into account the effects of the dry friction. Differently from the second order models, the first order ones are in this case absolutely inadequate to the purpose.
Figure 9 refers to an input sinusoidal command (frequency 2 Hz, amplitude 0.001 m) with low flow gain. As a consequence of the low input amplitude, no saturation is involved in the dynamic response of the system, therefore the responses of all the models substantially fit XJ.
Figure 10 refers to an input sinusoidal command (frequency 2 Hz, amplitude 0.01 m) with low flow gain. As a consequence of the large input amplitude, the saturations concerning force and actuation rate are involved in the dynamic response of the system, so the first order (XJ1S) and the second order (XJ2S) non linear models are characterised by a sequence of ramp re

sponses (marked delayed phase angle and reduced amplitude) sufficiently similar to the reference model ones XJ; on the contrary the first order (XJ1L) and the second order (XJ2L) linear models show a different trend, closer to the input command with low delay and no amplitude reduction. Differently from the non linear models, the linear ones are in this case inadequate to the purpose.

Figure 11 refers to an input step command from 0 to 0.003 m with low flow gain and step load of 8000 N. The responses of the linear models are too fast because they are not able to take into account the maximum value of the actuation speed of the system; the first order models, owing to their insensitivity to the load, are not able to represent the correct piston displacement. The only second order non linear model XJ2S is sufficiently able to fit the reference model XJ, except for the large overshoot in the return travel of the reference model due to the aiding load, not reproducible by the proposed model.

Figure 12 refers to an input step command from 0 to 0.0003 m with high flow gain and no load. The high flow gain gives a low stability margin to the system, so some overshoots are present in the response; the only reference model, having the ability to compute the effect of the dry friction, shows a consequent steady state position (with error) at the end of the actuation travel. No difference is shown between the displacements of the linear models and the corresponding non linear ones, because the small actuation travel does not involve any saturation. Obviously, at the beginning of the actuation travel, the first order models start move quickly, followed by the second order ones, later followed by the reference model. Generally the proposed models are able to represent with an acceptable accuracy the actual system behaviour (nevertheless the higher overshoots).

Figure 13 refers to an input step command from 0 to 0.01 m with high flow gain and no load.

With respect to the reference model (XJ), the second order models (XJ2L and XJ2S) show a lower overshoot (intrinsically not present in the responses of the first order models).

The responses of the linear models are too fast because they are not able to take into account the maximum value of the actuation speed of the system. The first order non linear model XJ1S shows a response in advance with respect to the reference model while the non linear second order model XJ2S sufficiently fits XJ (nevertheless the smaller overshoots). Generally the non linear models are able to represent with an acceptable accuracy the actual system behaviour, while the linear ones are markedly inadequate.

5 Conclusions

As a general result, the following considerations can be done: both the first order linear model and the second order linear one are therefore equally able to highlight the position error caused by the presence of an actuation rate DXJ in steady state; the second order models, unlike the first order ones, are also able to highlight the effect of the hinge moment on the aerodynamic surface (both in terms of steady state position error and of altered actuation rate), the possible oscillations of the servomechanism and the possible overshoots in its responses; furthermore, both the first order non-linear model and the second order non-linear one are equally able to highlight when the aerodynamic surface reaches its limit stops (results not presented); nearly the same can be said about reaching the maximum actuation rate; the second order non-linear model, unlike the first order one, is also able to highlight the achievement of the maximum drive torque in steady state conditions (stall) (results not presented).

As shown by the previous simulations, in case of small commands and low flow gain the behaviour of all the simplified models sufficiently fits the reference one; in case of small commands and high flow gain only the behaviour of the second order models (XJ2L, XJ2S) is satisfactory. In case of large commands and low flow gain only the non linear models (XJ1S, XJ2S) are generally able to give a sufficiently accurate response (if a load acts on the moving surface, the second order model only is sufficiently accurate); in case of large commands and high flow gain only the non linear second order model XJ2S sufficiently fits the reference model .

As a consequence, the results point out that only the proposed non linear second order model XJ2S is a satisfying answer to the demand of simple, but acceptably accurate, simulation model of the flight controls as a part of the complete simulation of the aeroplane – autopilot – servocontrol system (in order to maintain within acceptable limits the system complexity).

6 References

[1] G. Jacazio, L. Borello - "Mathematical models of electrohydraulic servovalves for fly-by-wire flight control systems" - Mathematical computing modelling, Vol. II, pp. 563-569, 1988

[2] Jacazio G., Borello L. - A non-linear model of an electrohydraulic servo system with axial piston hydraulic motor - 7th International Fluid Power Symposium, 16-18 September 1986, Bath - England.

[3] Jacazio G., Borello L.- Mathematical models of electrohydraulic servovalves for fly-by-wire flight control systems - 6th International Congress on Mathematical Modelling, August 1987, St.Louis – Missouri – U.S.A.

[4] Jacazio G., Borello L., Villero G. - Secondary flight controls: mechanical failures of transmission lines and related asymmetry problems - 19th ICAS Congress, Anaheim, California, U.S.A., 18-23 September,1994.

[5] Borello L., Villero G. - An innovative architecture for flap control system of medium/large aircrafts - 1st AIAA Aircraft Engineering, Technology, and Operations Congress - September 19-21, 1995, Los Angeles, CA.

[6] Borello L., Villero G. - Mechanical failures of flap control systems and related monitoring techniques - 21st ICAS Congress, 13-18 September 1998, Melbourne, Australia.

7 List of symbols

AJ	Hydraulic piston area
ASV	Servovalve second stage area
CLJ	Viscous damping coefficient (linear)
CSJ	Viscous damping coefficient (non linear)
CJ	Viscous damping coefficient
Com	Command
Cor	Servovalve current
CorM	Servovalve saturation current (module)
Err	Position error
D2XJ	Hydraulic piston acceleration
DePC	Differential pressure acting on the hydraulic piston at null flow
DePM	Saturation value of DePC (module)
DePJ	Actual differential pressure acting on the hydraulic piston
DXJ	Hydraulic piston rate
FM	Hydraulic piston motive force
FMM	Saturation value of FM (module)
FR	External load acting on the hydraulic piston – surface assembly
GAP	Position loop proportional gain (reference model)
GAS	Rate loop gain
GM	Servovalve first stage torque motor gain
GPS	Servovalve second stage pressure gain
GPSS	Servovalve second stage pressure gain in saturation conditions
GQF	Servovalve first stage flow gain
GQS	Servovalve second stage flow gain
KF	Servovalve first stage spring stiffness
KJ	Position loop proportional gain (simplified models)
KSF	Servovalve first – second stage spring stiffness
MJ	Hydraulic piston and surface mass
QJ	Hydraulic piston flow
XF	Servovalve first stage displacement
XFM	Servovalve first stage end of travel (module)
XJ	Hydraulic piston displacement (reference model)
XJ1L	Hydraulic piston displacement (linear first order model)
XJ1S	Hydraulic piston displacement (non-linear first order model)
XJ2L	Hydraulic piston displacement (linear second order model)
XJ2S	Hydraulic piston displacement (non-linear second order model)
XJM	Hydraulic piston end of travel for all the models (module)
XS	Servovalve second stage displacement
XSM	Servovalve second stage end of travel (module)

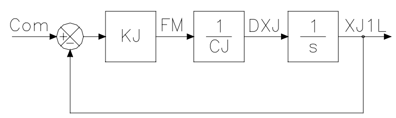

Fig. 1 – Linear first order model insensitive to the load

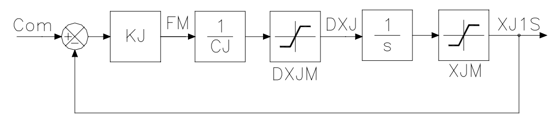

Fig. 2 – Non–linear (saturation) first order model insensitive to the load

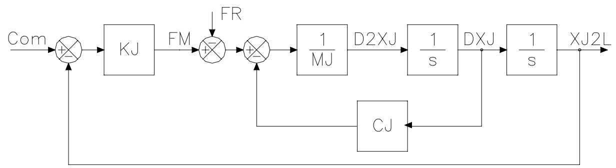

Fig. 3 – Linear second order model sensitive to the load

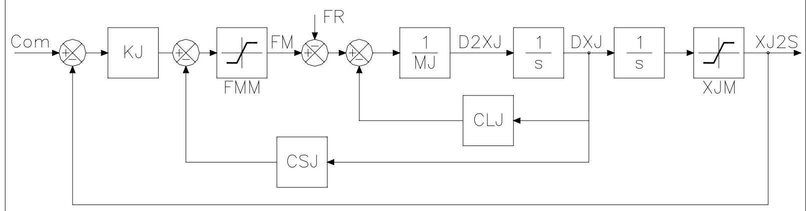

Fig. 4 – Non–linear (saturation) second order model sensitive to the load

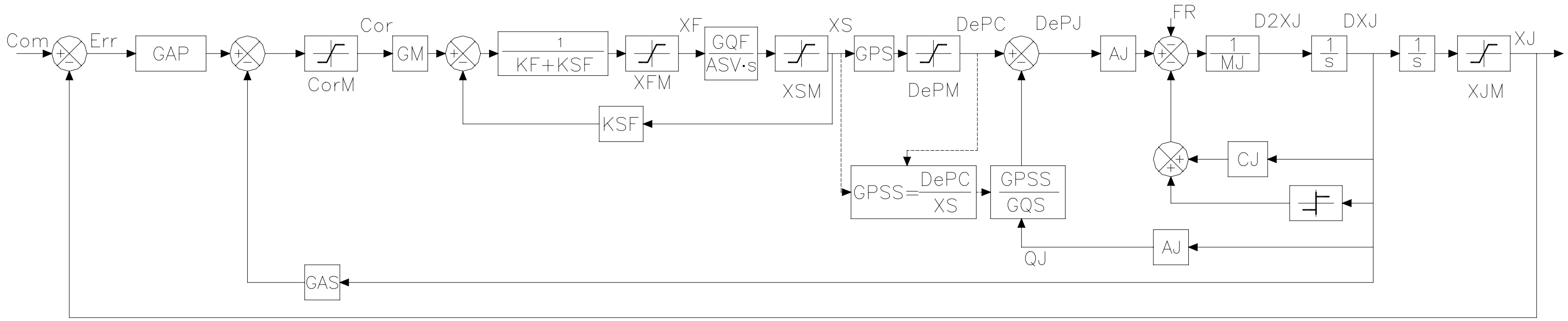

Fig. 5 – High order reference model

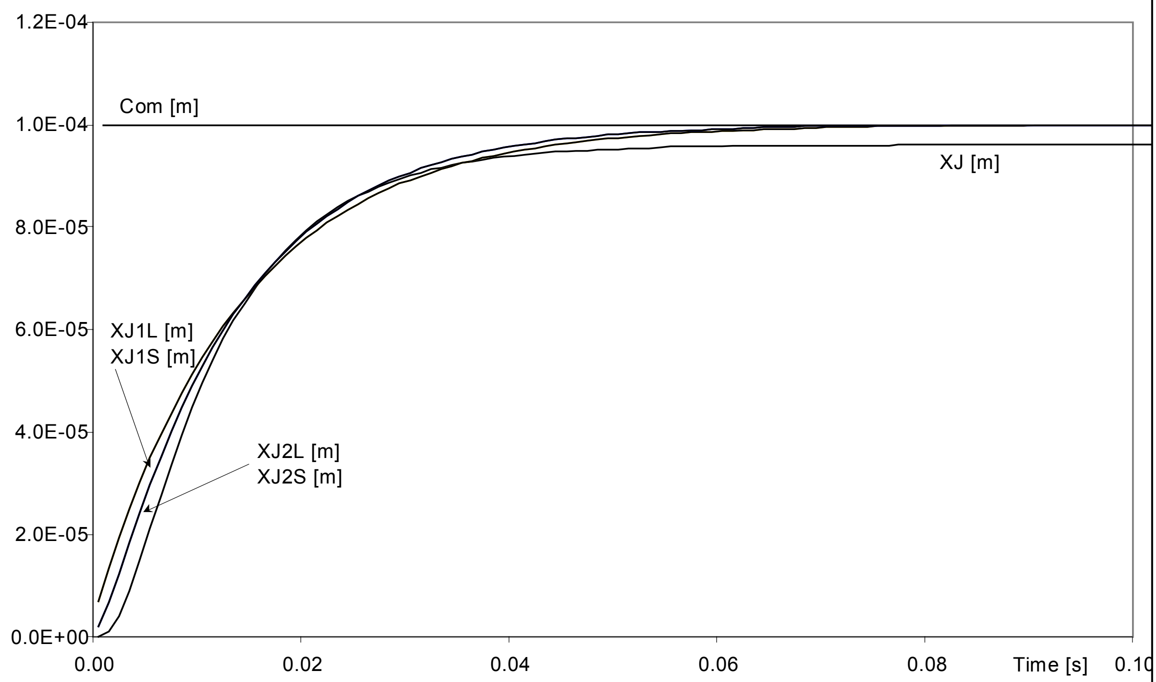

Fig. 6 - Small step command with low flow gain (no load)

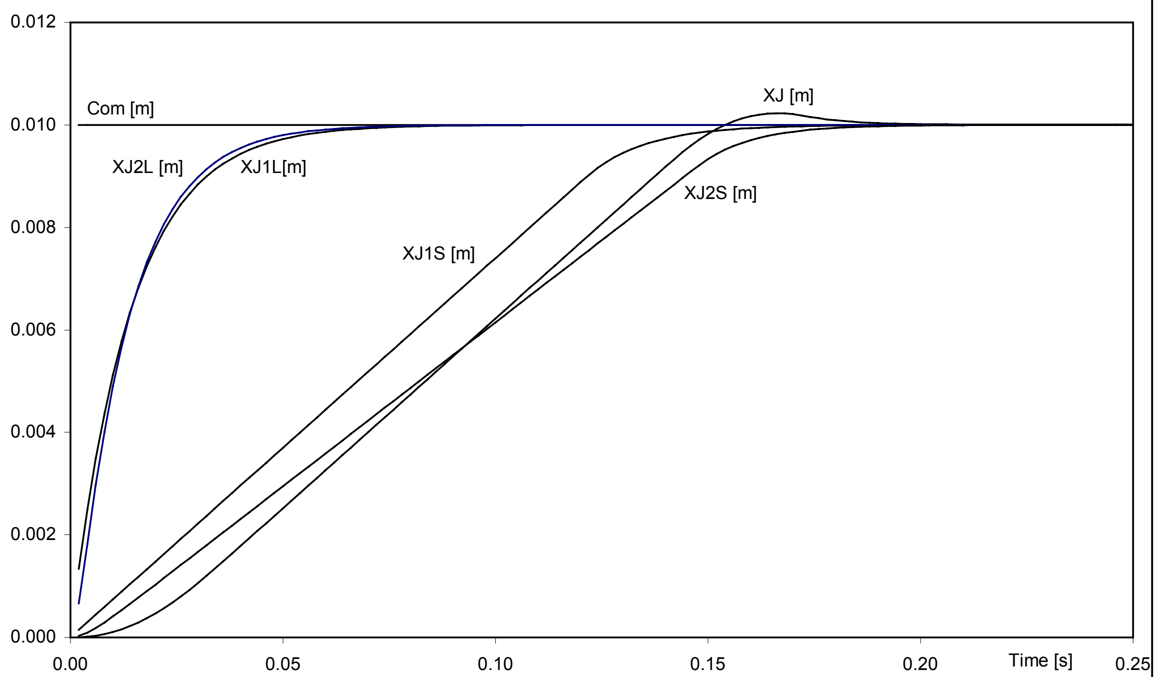

Fig. 7 - Large step command with low flow gain (no load)

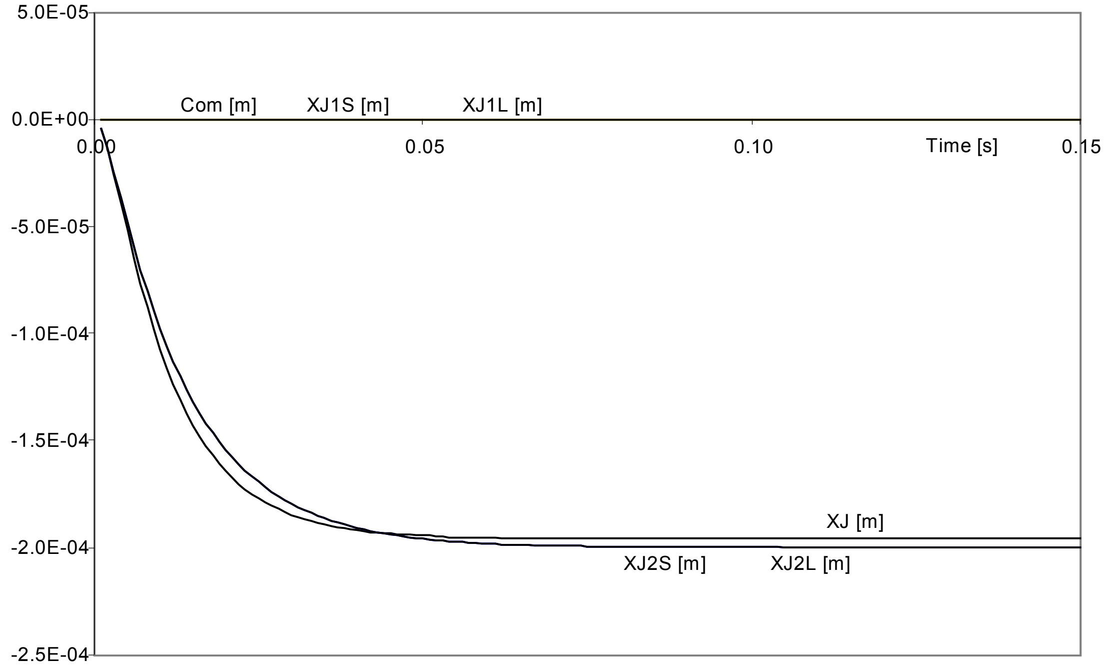

Fig. 8 - Step load with low flow gain (no command)

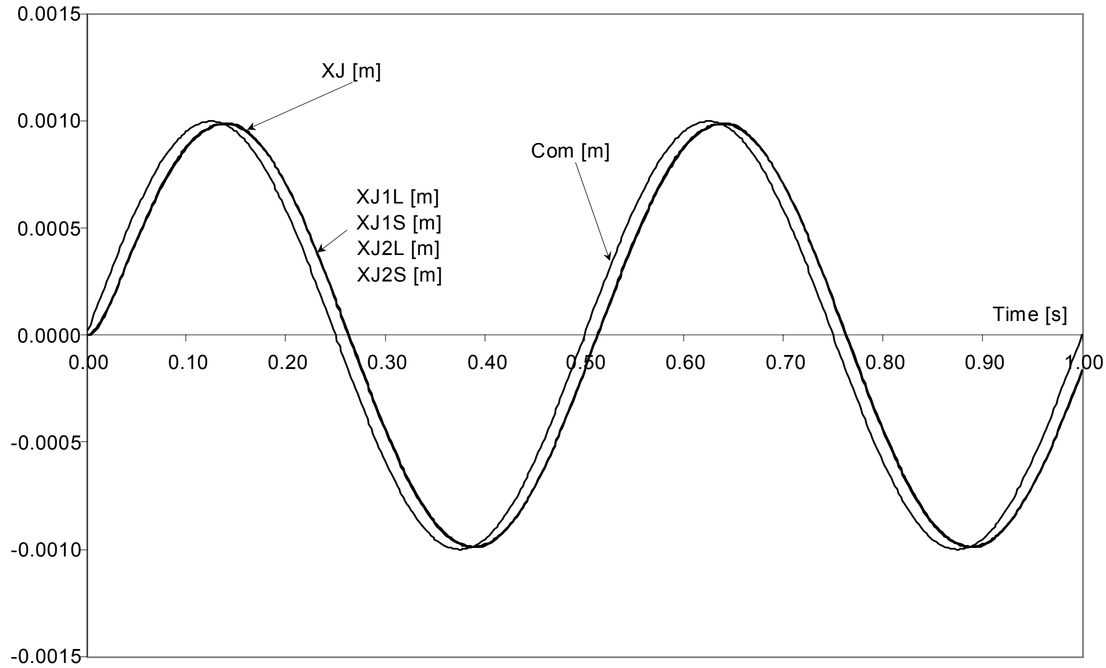

Fig. 9 - Sinusoidal command (low amplitude) with low flow gain (no load)

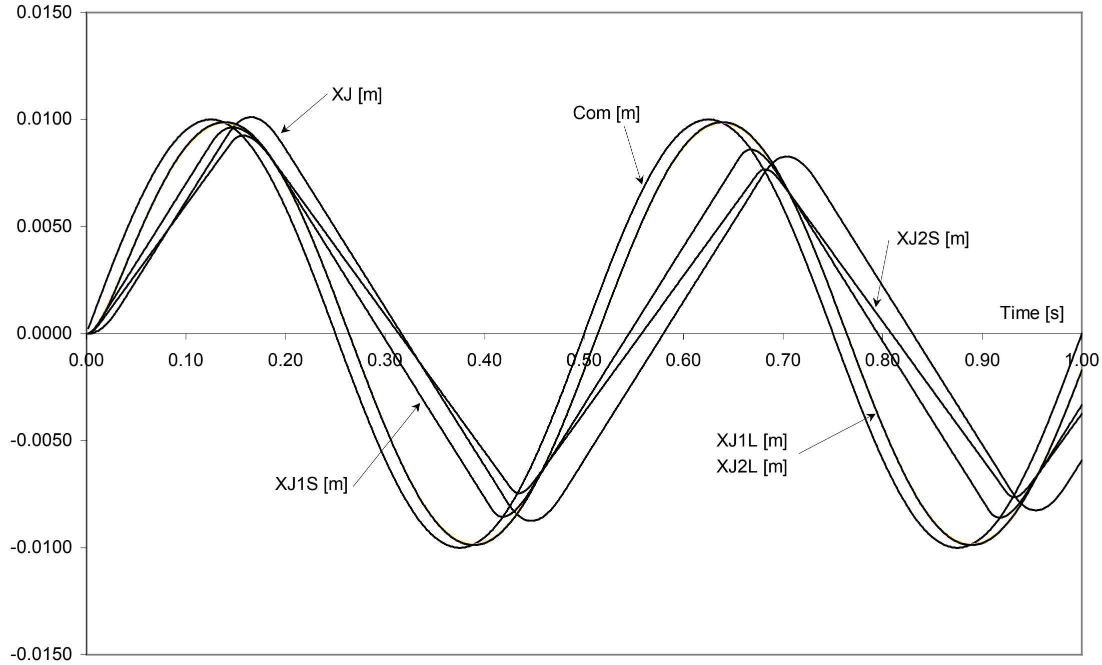

Fig. 10 - Sinusoidal command (large amplitude) with low flow gain (no load)

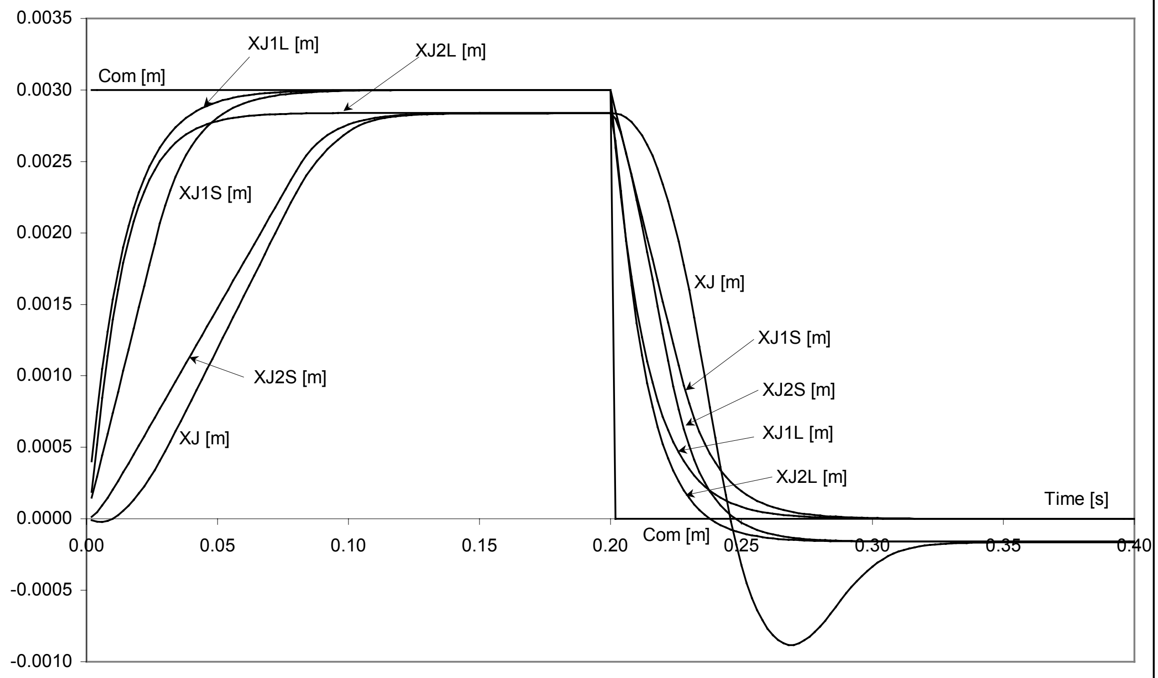

Fig. 11 - Large step forward command and rearward with low flow gain and load

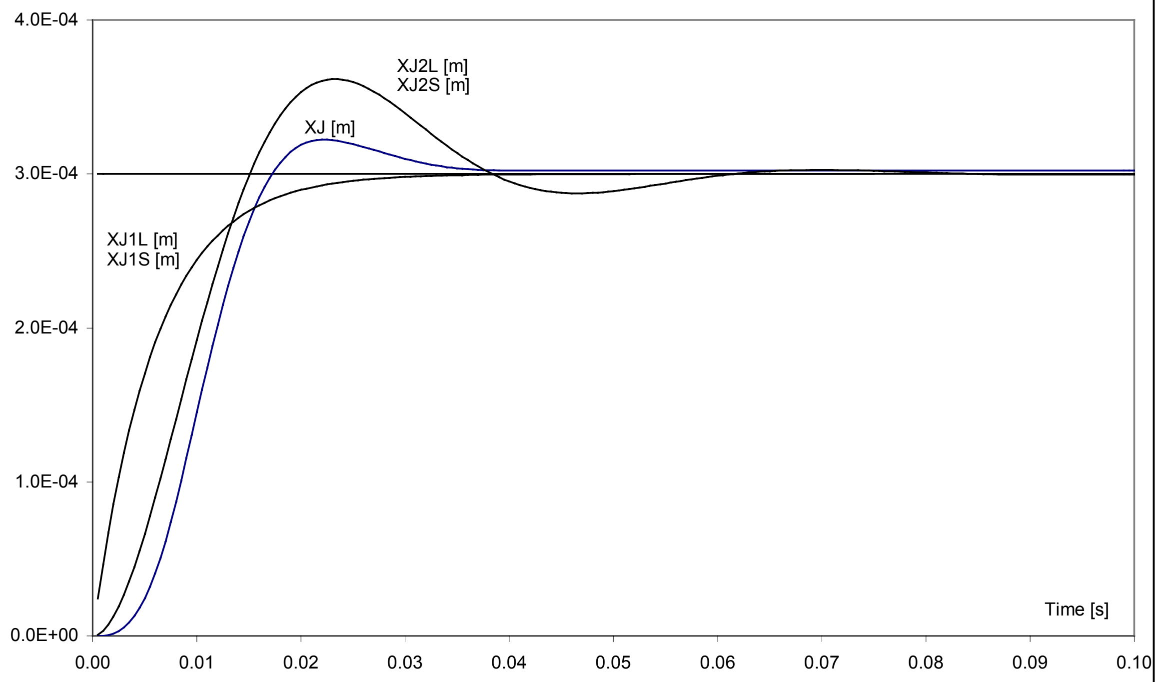

Fig. 12 - Small step command with high flow gain (no load)

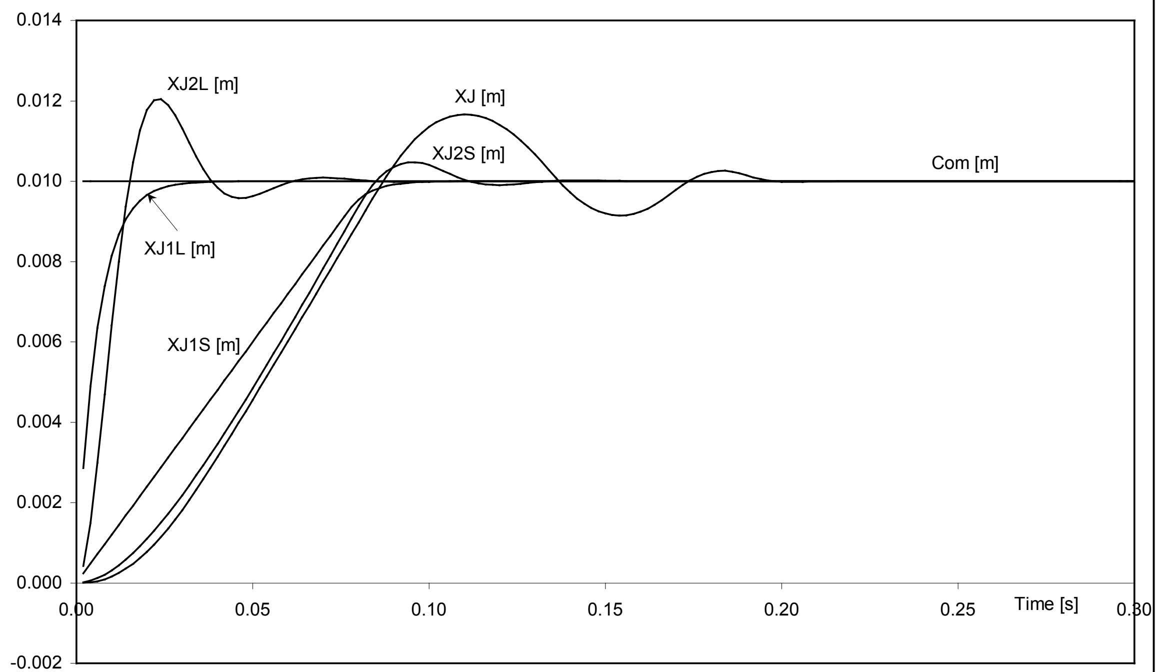

Fig. 13 - Large step command with high flow gain (no load)