A Numerical and Experimental Fluid-dynamic Analysis of a Hydraulic Actuator by means of Closed Loop Tests

M. Cardone, S. Strano, M. Terzo, and G. Vorraro

Abstract— **A numerical and experimental analysis of a hydraulic actuation system is described in this paper. The approach is based on a combined use of measurements and simulations. The model has been developed adopting a commercial code that allows to take several details into account, differently from a typical mathematical model. The model has been derived taking into account all the components of the hydraulic circuit. The results of numerical simulations are reported and compared with experimental data to show the validation and the performances of the developed model.**

Index Terms— **AMESim simulation, closed loop test, Hydraulic system, system characterization, fluid-dynamics.**

I. Introduction

THE employment of the hydraulic actuation for the positioning/loading systems needs of suitable control laws in order to make effective the several purposes. As a consequence, the development of a reliable model of the system has to be considered as a starting step of the controller design.

The hydraulic actuation system under consideration is characterized by a high power/mass ratio and a fast response. At the same time, it exhibits significant non-linear behaviour due to the pressure-flow rate relationship, the dead zone of the control valve and frictions; these non-linearities make the mathematical model more complex and, at the same time, highly limit the performance achieved by the classical linear controller [1 - 4].

In order to obtain a virtual model which is more reliable, software packages are often used to model and analyze multi-domain systems and to predict their performances. This means that it allows to link different physics domains (hydraulic, pneumatic, mechanic, electrical, thermal, electromechanical). Model components are described using validated analytical models that represent the actual hydraulic, pneumatic, electric or mechanical behaviour of the system. The user can compose a physics-based model using sub-models that have to be linked.

One of multi-domain software commercial codes is LMS Imagine.Lab AMESim Suite (or AMESim); in literature there are many research papers concerning the utilization of AMESim in engineering applications [5 - 9]. In this paper, a numerical model of a hydraulic actuator is developed by means of the commercial code AMESim and, successively, validated by means of closed loop tests. The model takes into account all the components of the hydraulic circuit: the axial piston pump, the pressure relief valve, the main control valve, the accumulators, the hydraulic cylinder with variable displacement and all the connecting pipes. Particular attention has been focused on the modelling of the internal resistances of the hydraulic system and on the valve dead zone.

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II. Test rig description

The considered hydraulic actuation system is employed to carry out seismic testing (Fig. 1) [10] and the hydraulic circuit shown in Fig. 2 consists of an axial volumetric piston pump powered by a 57 kW electric motor.

Fig. 1. Test rig

The pump is characterized by a maximum flow rate equals to 313 l/min. The other three main parts of the hydraulic circuit are the four-way three-position proportional valve, the flow distribution system (that will be described in detail below) and the hydraulic cylinder. A pressure relief valve is located downstream of the pump.

The horizontal hydraulic cylinder is constituted by a cylindrical barrel divided into two equal parts by a diaphragm; inside each part there is a piston whose rod is connected to the fixed base; so, the actuator has a mobile

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barrel and fixed pistons. The four feeding chambers (1A, 2A, 1B, 2B), that are supplied through holes drilled along the axis of the rods, are shown in Fig. 3.

The flow distribution system allows to have a large operation field. In fact, through a system of six three-way valves and two servo-valves it is possible to have different power configurations.

With reference to the sketch of Figure 3, the possible power configurations are:

a) 1A + 2B (or 1B +2 A): condition of maximum load and minimum speed;
b) 2A + 1A (or 1B +2 B): condition of minimum load and maximum speed;
c) 1A (or 1B): intermediate state;
d) 2A (or 2B): intermediate state.

In this way it is possible to obtain different hydraulic cylinder load areas. Hence, varying the power configuration it is possible to obtain, for example, different values of the table velocity using the same value of the hydraulic cylinder input flow rate, regulated by the proportional valve. The load areas corresponding to different power configurations are reported in Table 1.

TABLE I
HYDRAULIC CYLINDER LOAD AREA FOR DIFFERENT POWER CONFIGURATIONS

Power configuration	**Hydraulic cylinder load area**	**Value**	**Unit**
(a)	A_1	89.69	cm^2
(b)	A_2	23.75	cm^2
(c)	A_3	56.72	cm^2
(d)	A_4	32.97	cm^2

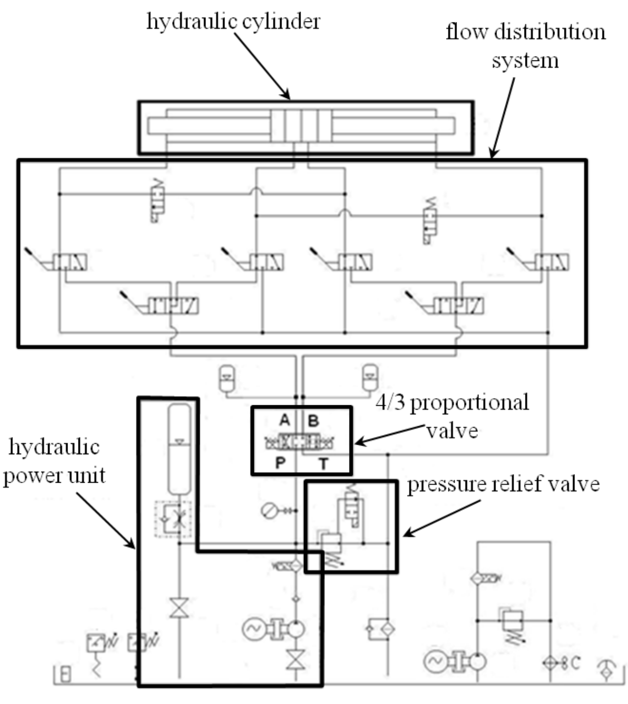

Fig. 2. Hydraulic circuit.

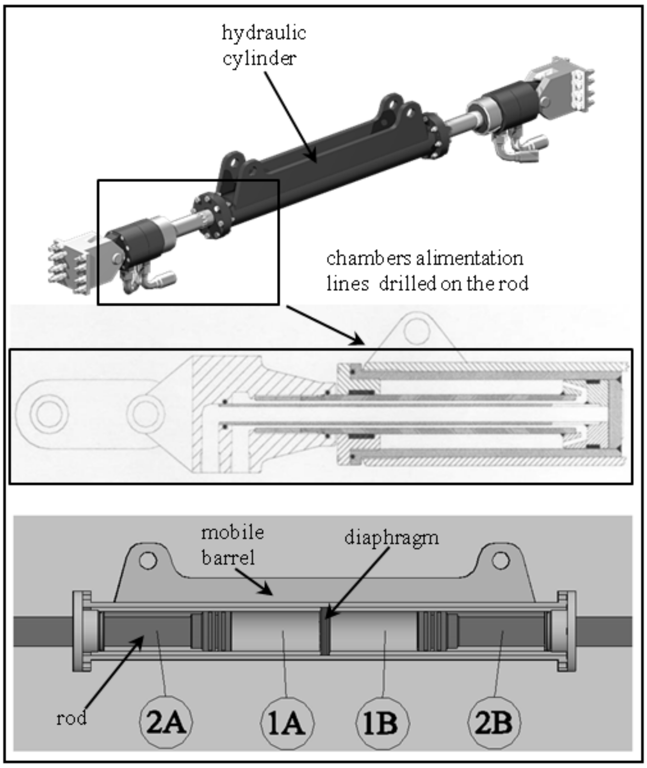

Fig. 3. Hydraulic cylinder details.

The position of the mobile barrel and the force exerted by the actuator on it are detected by a position sensor and a load cell respectively. Fig. 4 shows the force couple (F·h) acting on the whole system (hydraulic cylinder + shaking table), caused by the action of the cylinder and the reaction of the device under test, is balanced by the vertical reactions of the linear guides (R·d).

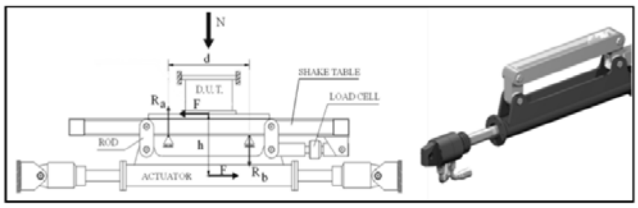

Fig. 4. Reaction forces of the linear guides.

The maximum horizontal force is 190 kN, the maximum speed 2.2 m/s and the maximum stroke is 0.4 m (± 0.2 m).

In addition to the actuator displacement and force sensors the following measurements are used:

- pressure in P, T, A and B port of proportional valve;
- proportional valve spool position.

III. Fluid Dynamic Model

The fluid-dynamic analysis of the hydraulic actuator has been performed by means of the commercial code LMS Imagine.Lab AMESim Suite.

Fig. 5 shows a scheme of the simulation model in AMESim environment. In this figure it is possible to observe the hydraulic components of circuit and the control systems.

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The model has been checked step by step; the first step has been the making and the validation of the sub-models. Then all the sub-model components have been assembled to realize the complete model of the hydraulic circuit.

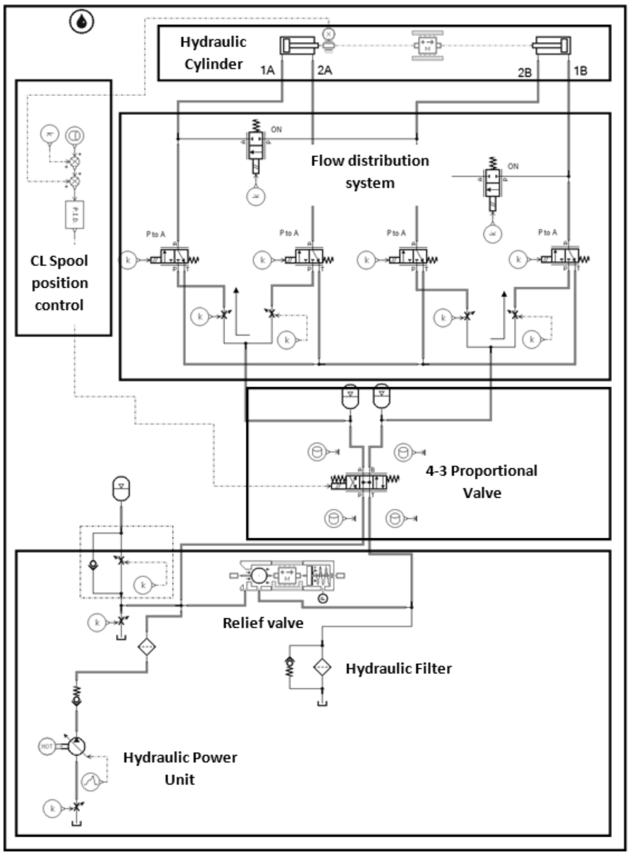

Fig. 5. Sketch of the simulation model.

The oil temperature is kept constant by a heat exchanger; for this reason, the AMESim hydraulic circuit model does not take into account the temperature influence on the system behaviour.

In Fig. 5 it is possible to see the architecture of the test bench hydraulic circuit: the power unit (with pressure pump, relief valve, accumulator, etc.), the 4-3 proportional valve, flow distribution system, hydraulic cylinder, etc. After the realization, the simulation model has been tuned and validated; in particular it has been validated first each component and then the overall model.

Validation of the components has been achieved by starting from the experimental data available by manufacturers [11], while the validation of the circuit model has been obtained by comparing simulated data with the experimental ones acquired on the test bench.

For example, the simulated characteristic curves of the two hydraulic components are shown in Fig. 6 and 7: the relief valve and the flow control valve.

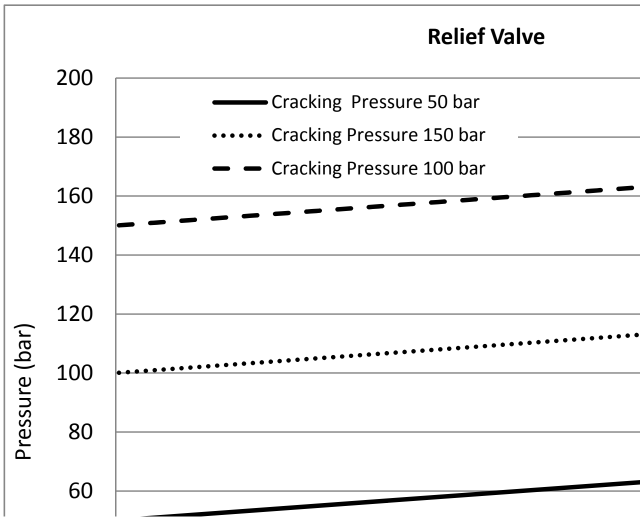

Fig. 6. Relief valve characteristics.

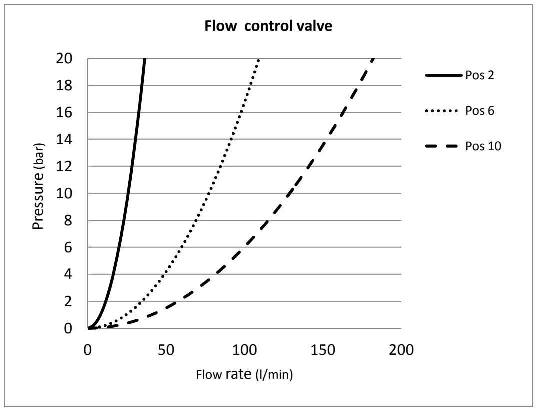

Fig. 7. Flow control valve characteristics.

After the model building up and its geometrical set-up, an accurate tuning and validation have been carried out by a great amount of experimental data measured on test bench by dedicated data acquisition system.

The oil pressure at the four ports (P, A, B and T of Fig. 8), the spool position of proportional valve and the hydraulic cylinder position have been acquired.

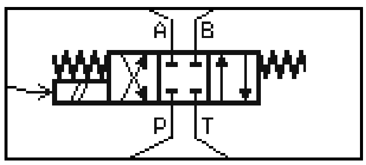

Fig. 8. Valve scheme.

The experimental tests have been carried out in closed loop with different settings of oil pressure and spool position [12]. All the tests have been conducted choosing the power configuration corresponding to the maximum load and minimum speed (condition (a) in Table I).

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The model calibration has involved the optimises the discharge coefficients of all the hydraulic components (valves, pipes, cylinders, etc.) and in particular the characterization of the dead zone of the proportional valve.

The influence of dead zone on the performance of the whole circuit is very sensitive to low values of the valve spool position, for this reason the model validation has been performed only under these conditions.

The model can be used to estimate the pressure drops in all hydraulic circuit points and the flow rates of all circuit components. For example, to estimate the flow rates in some critical zone that couldn't be measured experimentally.

As an example, a closed loop test, characterized by a supply pressure of 90 bar, will be discussed below. It has been realized taking into account a sinusoidal law for the target displacement (amplitude 0.1 m and frequency 0.5 Hz) and adopting a proportional feedback controller. In the following, the comparisons between the experimental and the simulated data are illustrated. They concern the control action, the actuator displacement and the oil pressure in P, A and B port.

Comparing all the simulation results with the experimental ones, it is possible to see an initial time interval (about 0.25 s) where the model is not able to predict the experimental results; this is due to unmodelled phenomena occurring in the initial transient state.

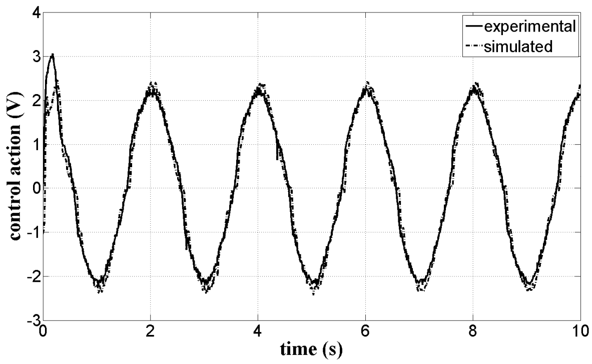

Fig. 9. Control action.

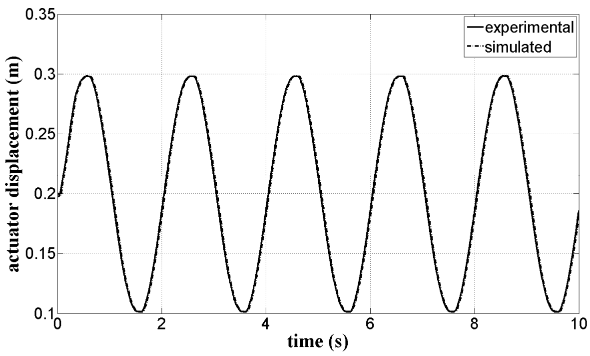

Fig. 10. Actuator displacement.

Adopting the same proportional gain of the feedback controller, the experimental and the simulated data in terms of control action (Fig. 9) and actuator displacement (Fig. 10) are practically superimposed and highlight the goodness of the numerical model for both the static and the dynamic contribution.

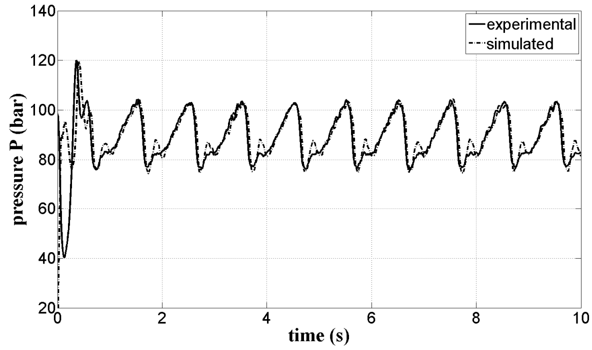

Fig. 11. Pressure at the port P.

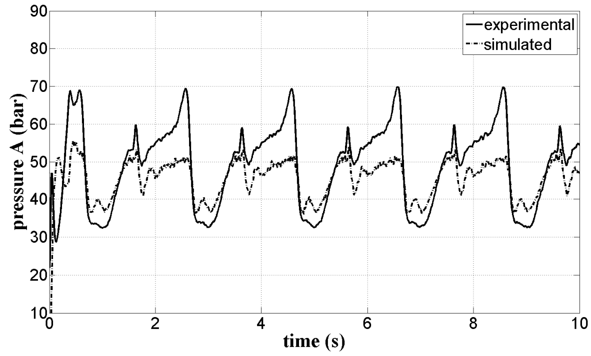

Fig. 12. Pressure at the port A.

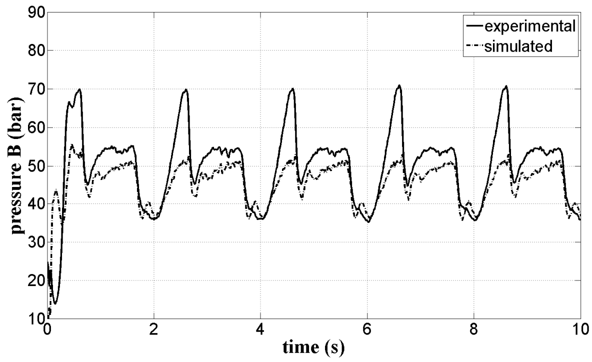

Fig. 13. Pressure at the port B.

Moreover, Fig. 11 shows a good prediction of the pressure in the port P.

The simulated pressures at the ports A and B (Figs. 12 and 13) are slightly different from the experimental ones; this is due to the difficulty of modelling the real fluid leakage and friction when the spool valve is positioned in correspondence of the dead zone.

Finally, the comparison between simulated and experimental results validate the modelling of the pipelines, relief valve and the proportional valve making the whole model a powerful tool for both the open and closed loop tests of the hydraulic actuator.

IV. Conclusion

A numerical and experimental investigation has been performed on a hydraulic actuator. The hydraulic model has been calibrated and validated by comparing experimental data, found during the several tests in closed loop mode, with the simulated ones.

The model is able to correctly predict the dynamics of the table and the fluid-dynamics of the hydraulic circuit.

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By using the model in co-simulation with control software developed in another simulation environment or in closed loop with real physical controller (hardware in the loop), it is possible to verify different control algorithms avoiding making tests on the real machine. Moreover, it is possible to predict the system response in the case of a mechanical, hydraulic or electrical component modification.

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