A Hydraulic Cylinder with Variable Power Configuration: Design and Validation of the Fluid-Dynamic Model

C. Abagnale and M. Cardone

Abstract— The paper describes the study of a hydraulic actuation system, in terms of design and validation of its fluid-dynamic model. A set of three way manual valves allows to select different cylinder thrust areas, making possible variable power configuration. The activity has been carried out through a combined use of measurements and simulations. The model of the hydraulic actuation system has been developed adopting a commercial code that allows to take several details into account, differently from a typical mathematical model: all the components of the hydraulic circuit have been taken into account for the derivation of the model. In order to characterize the system, the experimental activity has been conducted by open and closed loop tests. The results of numerical simulations have been compared with experimental data to evaluate the validation and the performance of the developed model.

Keywords— AMESim simulation, closed loop test, Hydraulic system, system characterization, fluid-dynamics.

I. Introduction

Hydraulic actuation systems are widely used in industrial applications due to their small size-to-power ratios and their ability for generating large actuation forces and torques at fast motion. Examples of hydraulic positioning systems can be found in transportations, industrial machineries, seismic applications [1-4]. However, the dynamics of hydraulic systems are highly nonlinear [5] due to the pressure-flow rate relationship, the dead band of the control valve and the frictions. In literature there are many studies concerning modelling and control of these devices; some of these focus on the modelling of the actuation system for the control development [6-8], others relate to the study of the influence of specimen under test on the dynamics of the shaking table and focus on the development of control systems that take into account such effects [9-10].

The aim of this study is to provide a virtual model of an experimental test bench suitable for vibration control system characterization [11]. The test rig mainly consists of a sliding table, driven by a hydraulic cylinder, on which the isolator under test is connected; the other end of the isolator is connected to a reaction structure. The peculiarity of the machine is that it may also have other purposes: it can be employed to deduce friction between sliding surfaces or, by disassembling the reaction structure, it can be adopted as a mono-axial shaking table.The machine has been designed to have displacement or force feedback controller. In this application it is important to design control systems that are insensitive to the unknown forces caused by devices under test (isolators or isolated structures) [12, 13]. In order to obtain a virtual model more reliable are often used software packages that offer simulation suite to model and analyze multi-domain systems and to predict their performance. 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, than for this purpose each sub-model has ports, which can have several inputs and outputs. One of multi-domain software commercial codes is LMS Imagine.Lab AMESim Suite (or AMESim); in literature there are a lot of research papers concerning the utilization of AMESim in engineering applications. For example in [14] it was used as a simulation experimental platform for design and optimization of the hydraulic system of the beam part of a tile press. In [15] AMESim was used to examine the performance of the developed control technique for electro-hydraulic valvetrains [16-18]. In this paper it is presented the AMESim hydraulic circuit fluid-dynamic model of the earthquake shaking table in order to have a virtual model of the machine much more faithful to the reality. It was analyzed in detail the hydraulic circuit; in fact the choice of AMESim components has been made on the basis of the datasheets provided by the manufacturers. The model takes into account all the components of the hydraulic circuit.

II. Hydraulic cylinder description

The hydraulic cylinder is constituted by two equal parts separated by a diaphragm and it contains two pistons which

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rods are connected to the base (Fig. 1); so, the actuator is characterized by a mobile barrel and fixed pistons. The maximum horizontal force is 190 kN, the maximum speed 2.2 m/s and the maximum stroke 0.4 m (± 0.2 m).

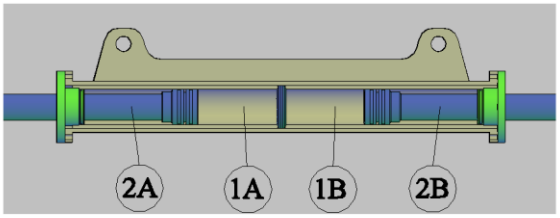

Fig. 1. Hydraulic cylinder detail.

With reference to Fig. 1, the chambers A and B can be divided in two sub-chambers respectively (1A, 2A, 1B, 2B). The two centred sub-chambers (1A and 1B) are supplied by means of suitable holes realized in the pistons, the other two sub-chambers (2A and 2B) are supplied directly by pipelines (Fig. 2) and the flow is regulated by means of a proportional valve.

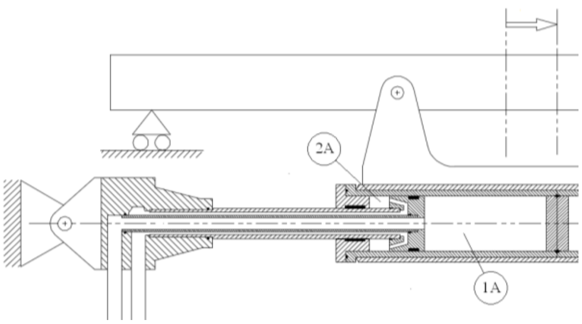

Fig. 2. Detail of the hydraulic circuit.

The hydraulic cylinder is employed in a test rig which can be use in two different configurations: seismic isolator testing system [19] and shaking table [20-22]. In the first configuration, the measurements can be useful to derive nonlinear models of the seismic isolator under test [23]; in the second configuration, dynamics properties of the base-isolated structure can be analysed [24, 25].

Fig. 3 shows that the hydraulic cylinder is linked between a fixed base and a sliding table.

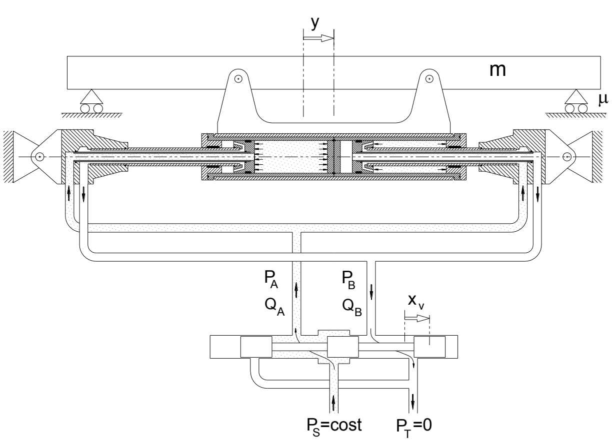

Fig. 3. Mounting configuration of the hydraulic cylinder.

Three way manual valves (see Fig. 4) allow to select different configurations of the test rig activating different thrust area (active surface) and control volume. Indeed the chambers can be supplied in four different ways:

1A + 2B (or rather 1B+2A);
1A + 2A (or rather 1B+2B);
1A (or rather 1B);
2A (or rather 2B).

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)	A1	89.69	cm2
(b)	A2	23.75	cm2
(c)	A3	56.72	cm2
(d)	A4	32.97	cm2

Condition a) represents the maximum thrust (minimum speed) configuration (called seismic isolator configuration) obtained increasing the active surface of the actuator and it is employed to perform shear tests on the seismic isolators that require greater forces to be horizontally strained. Condition b) is the maximum speed (minimum thrust) configuration (called shaking table configuration) reached by the reduction of the active surface and it is adopted in case of shaking table tests that need a higher frequency. Conditions c) and d) represent intermediate configurations.

The selection of the configuration a) and the installation of suitable reaction structures allow the bench to be employed as seismic isolator test rig; conversely, the removal of the reaction structures, together with the selection of the configuration b), allows the seismic test rig to be used as shaking table.

Fig. 4 shows the main parts of the hydraulic power unit. It consists of an axial volumetric piston pump powered by a 75 kW electric motor. The pump is characterized by a maximum flow of 313 l/min. The other three main parts of the hydraulic circuit are the four-way three-position proportional valve, the flow distribution system (with manual valves for the power configuration selection) and the hydraulic cylinder. A pressure relief valve is located downstream of the pump.

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 [26 - 29].

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.

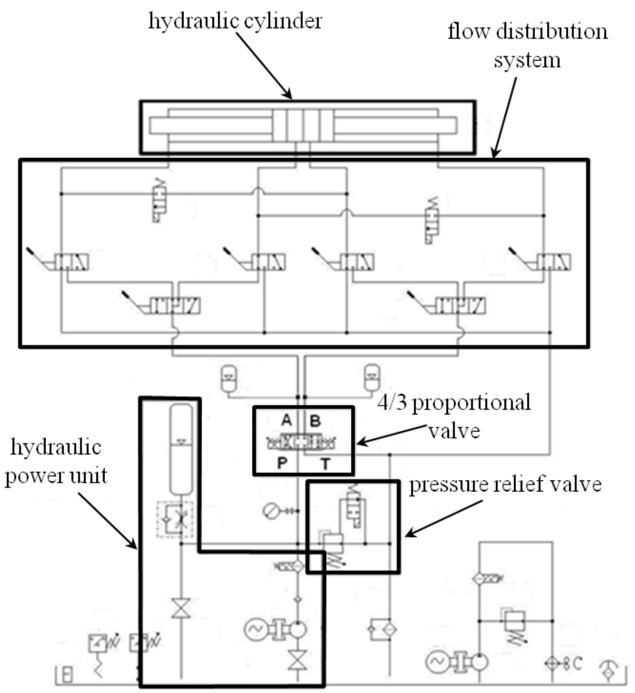

Fig. 4. Hydraulic circuit.

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.

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.

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, while the validation of the circuit model has been obtained by comparing simulated data with the experimental ones acquired on the test bench.

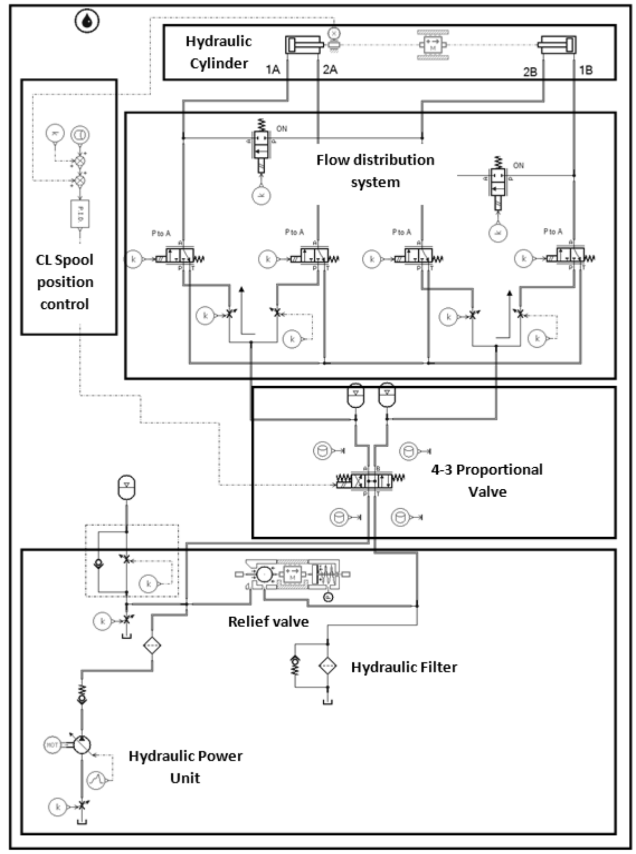

Fig. 5. Sketch of the simulation model.

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.

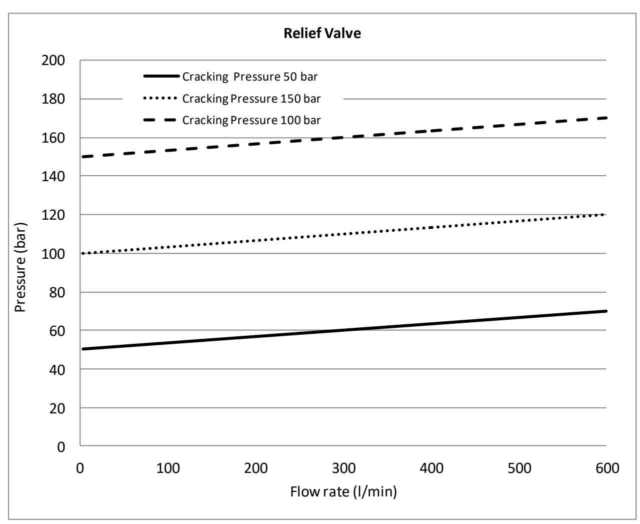

Fig. 6. Relief valve characteristics.

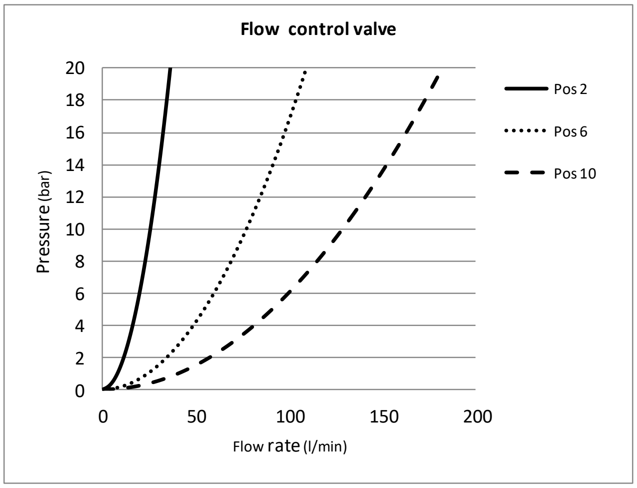

Fig. 7. Flow control valve characteristics.

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. 4), the spool position of proportional valve and the hydraulic cylinder position have been acquired.

IV. Model Validation in Open Loop Configuration

The experimental test for the model validation has been carried out in open loop mode choosing the power configuration corresponding to the maximum load and minimum speed (condition (a) in Table 1) [30].

The results of the test are shown in Fig. 8. The test was conducted at the assumed minimum value of the oil pressure (regulated by the relief valve) equal to 30 bar and choosing small displacements of the proportional valve spool. The assigned variable valve spool position signal was of step type with values ±10 % of its maximum displacement. In Fig. 8, are shown the experimental valve spool position and the comparisons between experimental and simulated data concerning oil pressure in P, A and B and table (or hydraulic cylinder) position during the test.

During the test the cylinder moves between the end positions several times. If the control signal (spool position) is positive, port P is connected to A and B is connected to T. At the end stop position, the pressure at port A is similar to that in P while B is similar to T. Of course, when the signal is reversed, the same considerations can be made by reversing the port A with B.

The dynamic pressure behaviour in the circuit (overpressures when the end stop is reached) is determined mainly by relief valve characteristics.

It’s possible to note that the dynamic characteristic of the utilized relief valve is very slow.

In the Fig. 8 it is also possible to observe the speed of the cylinder (slope of the line) which depends on the spool valve position (oil flow rate) and supply pressure.

Analyzing the validation test results it is possible to observe the good agreement between experimental and simulated data.

The model is able to correctly simulate both the fluid-dynamics of the hydraulic circuit and the dynamic displacement of the table.

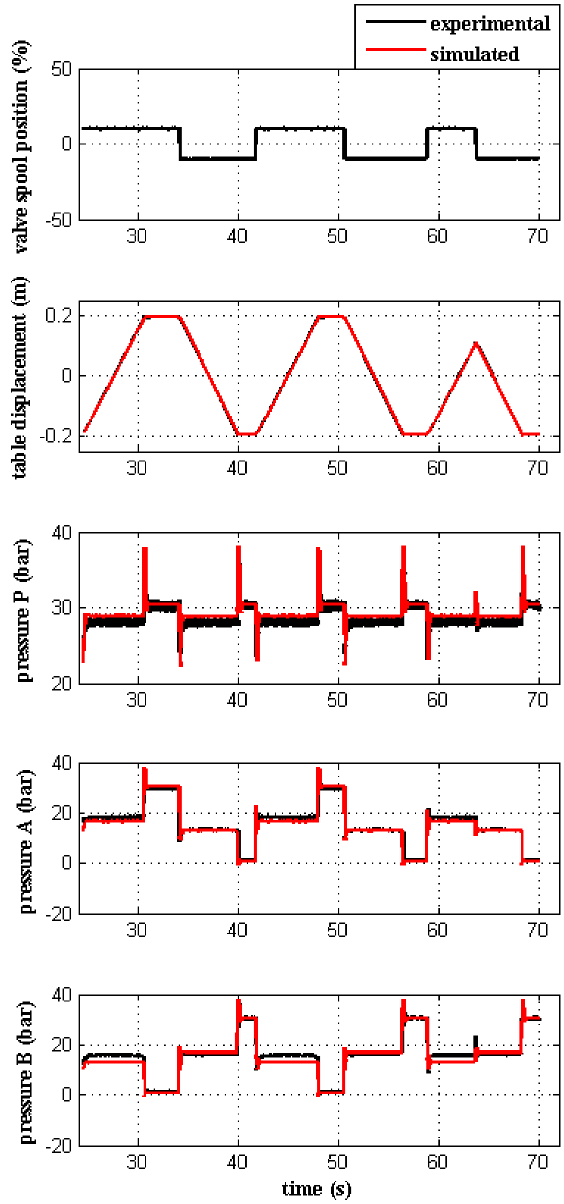

Fig. 8. Open loop test (supply pressure: 30 bar - valve spool position: ±10%).

V. Closed Loop Tests

A closed loop test, characterized by a supply pressure of 90 bar, 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.

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.

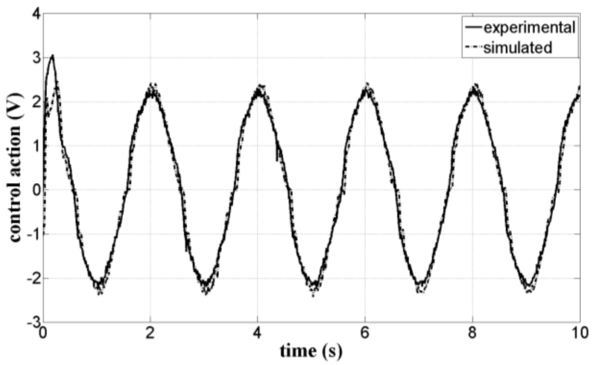

Fig. 9. Control action.

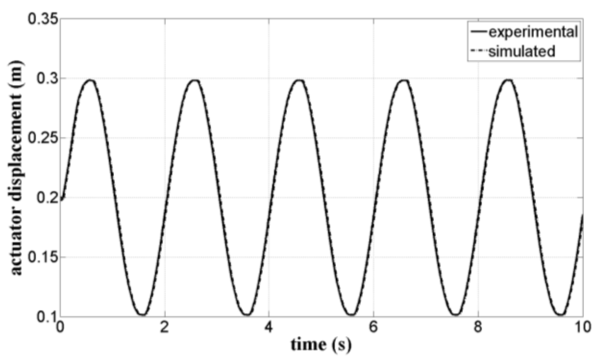

Fig. 10. Actuator displacement.

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.

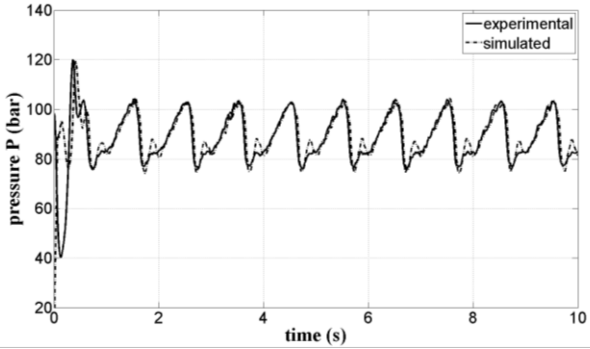

Fig. 11. Pressure at the port P.

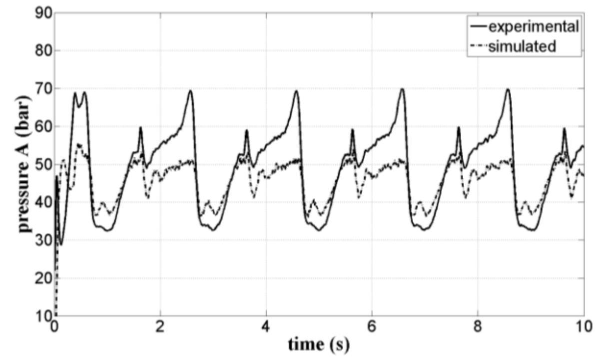

Fig. 12. Pressure at the port A.

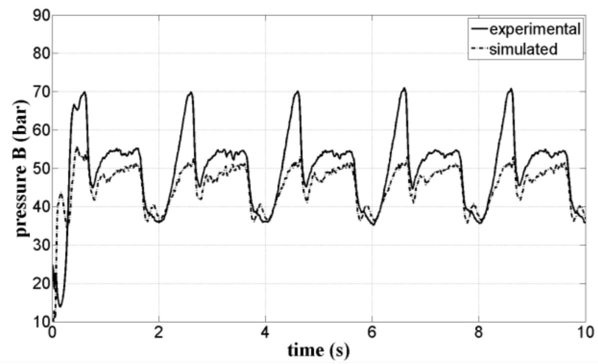

Fig. 13. Pressure at the port B.

VI. 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 open loop mode, with the simulated ones. Successively, closed loop tests have been performed in order to validate the control algorithm. The model calibration has involved the optimization of 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 model is able to correctly predict the dynamics of the table and the fluid-dynamics of the hydraulic circuit.

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

References

[1] H. P. Gavin, J. B. Hoagg, Control Objectives for Seismic Simulators, Proc. of the 2009 American Control Conference, St. Louis, MO, USA, 2009, pp. 3932-3937.

[2] J. P. Conte, T. L. Trombetti, Linear dynamic modeling of a uniaxial servo-hydraulic shaking table system, Earthquake Engineering and Structural Dynamics, vol. 29, no. 9, pp. 1375-1404, 2000.
[3] M. Blondet, C. Esparza, Analysis of shaking table-structure interaction effects during seismic simulation tests, Earthquake Engineering and Structural Dynamics, vol.16, no. 4, pp. 473-490, May 1988.
[4] S. Pagano, M. Russo, S. Strano, M. Terzo, Seismic isolator test rig control using high-fidelity non-linear dynamic system modeling, Meccanica, vol. 49, no. 1, pp. 169-179, 2014.
[5] H. E. Merritt, Hydraulic Control Systems. New York: Wiley, 1967.
[6] M. Cardone, S. Strano, M. Terzo, G. Vorraro, A numerical and experimental fluid-dynamic analysis of a hydraulic actuator by means of closed loop tests, Lecture Notes in Engineering and Computer Science: Proceedings of The World Congress on Engineering 2013, London, U.K., 3-5 July, pp. 2151-2155, 2013.
[7] S. Strano, M. Terzo, Accurate state estimation for a hydraulic actuator via a SDRE nonlinear filter, Mechanical Systems and Signal Processing, vol. 75, pp. 576-588, 2016.
[8] F. Liccardo, S. Strano, M. Terzo, Real-Time Nonlinear Optimal Control of a Hydraulic Actuator, Engineering Letters, vol. 21, no. 4, pp. 241-246, 2013.
[9] S. Strano, M. Terzo, A multi-purpose seismic test rig control via a sliding mode approach, Structural Control and Health Monitoring, vol. 21, no. 8, pp. 1193 – 1207, 2014.
[10] S. Strano, M. Terzo, A first order model based control of a hydraulic seismic isolator test rig, Engineering Letters, vol. 21, no. 2, pp. 52 – 60, 2013.
[11] A. Calabrese, G. Serino, S. Strano, M. Terzo, Experimental investigation of a low-cost elastomeric anti-seismic device using recycled rubber, Meccanica, vol. 50, no. 9, pp. 2201-2218, 2015.
[12] F. Liccardo, S. Strano, M. Terzo, Optimal control using state-dependent riccati equation (SDRE) for a hydraulic actuator, Lecture Notes in Engineering and Computer Science: Proceedings of The World Congress on Engineering 2013, London, U.K., 3-5 July, 2003-2007, 2013.
[13] S. Strano, M. Terzo, A non-linear robust control of a multi-purpose earthquake simulator, Lecture Notes in Engineering and Computer Science: Proceedings of The World Congress on Engineering 2013, London, U.K., 3-5 July, pp. 1687-1692, 2013.
[14] Xiao T, Jiang X L L, Wu B. Modeling and Simulation of Hydraulic System of the Beam Part of the Tile Press Based on AMESim. International Conference on Fluid Power and Mechatronics 2011, Beijing, China. 593-596.
[15] Pournazeri M, Khajepour A, Fazeli A. An Efficient Lift Control Technique in Electrohydraulic Camless Valvetrain Using Variable Speed Hydraulic Pump, SAE 2011 World Congress & Exhibition 2011, Detroit, MI, USA.
[16] C. Abagnale, M.C. Cameretti, L. De Simio, M. Gambino, S. Iannaccone, R. Tuccillo, Combined numerical-experimental study of dual fuel diesel engine, Energy Procedia , 45 (2014), pp. 721-730.
[17] C. Abagnale, M.no Migliaccio, O. Pennacchia, New mechanical variable valve actuation systems for motorcycle engines. Proc. IMechE, Part C: Journal of Mechanical Engineering Science , 229 (4) (2015), pp. 716–734.
[18] C. Abagnale, M.no Migliaccio, O. Pennacchia. Design of a New Mechanical Variable Valve Actuation system for Motorcycle Engines, Proc. of the 11th Biennal Conference on Engineering Design and Analysis, ASME, July 2-4, 2012, Nantes, France (ESDA2012-82317).
[19] A. Calabrese, G. Serino, S. Strano, M. Terzo, Investigation of the seismic performances of an FRBs base isolated steel frame through hybrid testing, Lecture Notes in Engineering and Computer Science: Proceedings of The World Congress on Engineering 2013, London, U.K., 3-5 July, pp. 1974-1978, 2013.
[20] A. Calabrese, S. Strano, M. Terzo, Real-Time Hybrid Simulations vs. Shaking Table Tests: case study of a fiber-reinforced bearings isolated building under seismic loading, Structural Control and Health Monitoring, vol. 22, no. 3, pp. 535–556, 2015.
[21] M. Cardone, S. Strano, Fluid-dynamic analysis of earthquake shaking table hydraulic circuit, ASME 2012 11th Biennial Conference on Engineering Systems Design and Analysis, ESDA 2012, 2, pp. 343-350, 2012.
[22] S. Strano, M. Terzo, A SDRE-based tracking control for a hydraulic actuation system, Mechanical Systems and Signal Processing, vol. 60, pp. 715-726, 2015.
[23] C. M. Chang, S. Strano, M. Terzo, Modelling of Hysteresis in Vibration Control Systems by means of the Bouc-Wen Model, Shock and Vibration, vol. 2016, Article ID 3424191, 14 pages, 2016. doi:10.1155/2016/3424191.
[24] A. Calabrese, S. Strano, M. Terzo, Parameter estimation method for damage detection in torsionally coupled base-isolated structures, Meccanica, DOI:10.1007/s11012-015-0257-2, 2015.
[25] A. Calabrese, G. Serino, S. Strano, M. Terzo, An extended Kalman Filter procedure for damage detection of base-isolated structures, EESMS 2014 - 2014 IEEE Workshop on Environmental, Energy and Structural Monitoring Systems, Proceedings, Naples; Italy; 17-18 Sep., pp. 40-45, 2014.
[26] C. Abagnale, M. Cardone, P. Iodice, S. Strano, M. Terzo, G. Vorraro, Analysis of a New Measurement System of the Chain Strength for Electrically Assisted Bicycles, Proc. of the ASME 12th Biennial Conference on Engineering Systems Design and Analysis (ESDA2014), ESDA2014-20364, Vol. 1, Copenhagen, Denmark, June 25 – 27, 2014.
[27] C. Abagnale, M. Cardone, P. Iodice, S. Strano, M. Terzo, G. Vorraro, Power requirements and environmental impact of a pedelec. A case study based on real-life applications, Environmental Impact Assessment Review, vol. 53, no. 7, pp. 1 – 7, 2015, doi:10.1016/j.eiar.2015.02.003.
[28] C. Abagnale, M. Cardone, P. Iodice, S. Strano, M. Terzo, G. Vorraro, Theoretical and Experimental Evaluation of a Chain Strength Measurement System for Pedelecs, Engineering Letters, vol. 22, no. 1, pp. 102 – 108, 2014.
[29] P. Iodice, C. Abagnale, M. Cardone, S. Strano, M. Terzo, G. Vorraro, Performance Evaluation and Environmental Analysis of an Electrically Assisted Bicycle Under Real Driving Conditions, Proc. of the ASME 12th Biennial Conference on Engineering Systems Design and Analysis (ESDA2014), ESDA2014-20438, Vol. 1, Copenhagen, Denmark, June 25–27, 2014.
[30] S. Pagano, R. Russo, S. Strano, M. Terzo, Non-linear modelling and optimal control of a hydraulically actuated seismic isolator test rig, Mechanical Systems and Signal Processing, vol. 35, no. 1 – 2, pp. 255 – 278, 2013.