An Hydraulic Test Rig for the Testing of Quarter Turn Valve Actuation Systems

Luca Pugi, Giovanni Pallini, Andrea Rindi, Nicola Lucchesi

1Dept. of Industrial Engineering
University of Florence
Florence, Italy

2MDM TEAM SRL
(Spin Off of Florence University)
Florence, Italy
Luca.Pugi@unifi.it

3VELAN ABV, SpA
Capannori (LU), Italy

Abstract - in this work, some preliminary design results concerning the development of a rig for the fast prototyping and the testing of the actuation system of large quarter turn valves for power plants are shown. In particular, friction forces generated on sealing and bushings by the internal pressurized fluid influence opening and closing torques of large valves. In order to test a valve actuation system, the proposed rig is able to simulate the resisting torque and more generally the mechanical impedance of pressurized quarter turn valve. The proposed actuation of the rig is hydraulic. In this work design and choosing of different solutions are discussed with a particular attention to robustness and stability troubles of the rig respect disturbances, limited bandwidth and interaction with partially unknown tested system

I. INTRODUCTION

There are several kind of process valves used in Oil&Gas industry and more generally for large process plants whose opening state is controlled by imposing a rotation of about 90° as example ball or butterfly valves. Angular position of this kind of valves are usually controlled by linear pneumatic or hydraulic actuators whose linear motion is converted in a corresponding rotation using simple transmission systems such as for example the Scotch-Yoke system represented in the scheme of Figure 1. Also different solutions including electric actuators and toothed transmission systems, like gears of transmission belts, should be found in literature where most of the available documentation should be referred to patents concerning industrial applications.

For fail safe applications, a desired fail-state of the valve is usually assured by the preload of linear spring as visible in the example of Figure 1.

The actuation system of the valve is usually controlled by a closed position loop implemented on a component briefly called as valve positioner or simply positioner. The positioner is mainly composed by the following components:

- Angular Position Feedback: angular position and consequently the state of the valve is continuously measured.
- Regulator: a simple controller typically a PID (Proportional, Integral, Derivative) position controller is implemented.
- Valve or Drive: according a reference command produced by the regulator, a Drive system, typically a Valve for pneumatic or hydraulic cylinders is used to finally control the actuator.

Figure 1. Example of Quarter Valve actuation system using a pneumatic cylinder and a scotch yoke transmission system

Aim of the proposed rig is to test the valve actuation system and positioner by simulating the opening/closing torque of a pressurized quarter turn valve. A simplified scheme is visible in Figure 2. The main advantage of this approach is to have the same actuation system running in different operating conditions and simulating different valves. This a quite interesting feature considering cost and encumbrances of valves and of the plant needed produce the pressurized flow in order to reproduce the typical operating conditions.

Tested valve actuator and positioner should be different both in terms of electro-mechanical layout and calibrations, as example in order to control different valves.
Therefore, there is the need to design the actuation and regulation of the rig to avoid potential troubles of stability due to the interaction with a tested system, whose behavior is only partially known.

More generally this problem should be treated as a particular case of a SISO (Single Input, Single Output) torque controlled system which have to be robust as much as possible respect to an external disturbance which should be treated as an imposed position/kinematics.

This kind of trouble has been widely studied in robotics for the following applications:

- force controlled or compliant end effectors [17]
- robots interacting with humans or potentially sensitive environments [14]-[16].
- walking robots were obstacles due to unstructured or partially unknown working environment should be treated as hard constraints or imposed position disturbances [18].

Similar problems are common on active or semi-active suspension systems, where the spectral content of the disturbances, that has to be rejected, should be far higher respect to the bandwidth of implemented control systems. Typical examples should be related to the control of railway pantograph with wire [19] or fluid [20] actuation systems. Also in the study of semi-active vehicular suspension, there is a robustness specification respect to a non-linear, bandwidth limited actuator [21].

Respect to previously cited applications, in this work attention is focused on an industrial application in which this kind of approach is quite uncommon. Furthermore, range of generated torque forces is quite rare respect to typical robotics or vehicular applications. Finally design has to be a compromise between cost, maintenance, ease of use specifications of an industrial applications against the necessity of performing a reliable Hardware in the Loop testing of positioners.

II. TESTING SPECIFICATIONS

Considering a wide variability of different positioners and actuators to be tested the specifications of the test rig are briefly summarized in: specifications in terms of maximum speed of actuators are quite not demanding considering that a complete opening or closing run require about 10 seconds.

More demanding requirements concerns the simulated torque profile: for the same simulated valve the value of the maximum resisting torque should be three times greater than the minimum one as visible in the example of Figure 3.

Considering that the ratio between two torques exerted by the biggest and the smallest tested actuator has a value of about ten, torque profiles that have to be correctly simulated by the rig actuator should be variable of more than thirty times.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Nom. VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angular Run [°]</td>
<td>80°-90°-100°</td>
</tr>
<tr>
<td>Max Angular Speed [rad/s]</td>
<td>0.137-0.143-0.157 [rad/s]</td>
</tr>
<tr>
<td>Max Acceleration [rad/s²]</td>
<td>0.08-0.144-0.28 [rad/s²]</td>
</tr>
<tr>
<td>Max Resisting Torque [Nm]</td>
<td>1000-5000-10000 [Nm]</td>
</tr>
<tr>
<td>Min. Resisting Torque [Nm]</td>
<td>350-1600-3300 [Nm]</td>
</tr>
</tbody>
</table>

III. DESIGN OF THE HYDRAULIC ACTUATOR

For the rig, an hydraulic actuation is preferred since it assure the possibility of exerting heavy torques with reduced encumbrances. Besides, it should be considered that the rig actuator has to be mostly regulated as brake able to dissipate a continuously controlled power. Management of electric drive is generally more complex in the braking phase, moreover it’s obvious to take into account penalties related to the limited quarter turn rotation and the impossibility of using gearings with high reduction ratio in order to reduce troubles related to friction, tooth backlash and friction.

Commercial hydraulic torque motors are available for sale in different size, however, considering cost specifications it was preferred a simple pivoted linear actuator linked to the rig using the transmission system described in the scheme of Figure 4.
An important advantage of this solution is the opportunity to scale the project depending on inputs: different solutions can be easily implemented to test bigger or smaller valves using commercial components. The main kinematic parameters of the transmission are described in TABLE II.

### TABLE II. MAIN GEOMETRIC PARAMETERS OF TRANSMISSION

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( R )</td>
<td>260[mm]</td>
</tr>
<tr>
<td>( x_p )</td>
<td>379[mm]</td>
</tr>
<tr>
<td>( y_c )</td>
<td>147[mm]</td>
</tr>
</tbody>
</table>

In particular solving this simple kinematics allows to calculate the ratio \( \tau(\alpha) \) between the exerted torque \( T \) and the corresponding linear force of the hydraulic cylinder, \( F \) (1).

\[
\tau = \frac{T}{F} = \frac{i}{\alpha} = \frac{r(x_p \cos \alpha + y_c \sin \alpha)}{\sqrt{(y_c - r \cos \alpha)^2 + (x_p + r \sin \alpha)^2}}
\]  

(1)

![Figure 4. Pivoted linear actuator connected to the rig using a simple rotoidal joint](image)

Figure 4. Pivoted linear actuator connected to the rig using a simple rotoidal joint.

Figure 5. shows the behavior of \( \tau(\alpha) \) normalized respect to its value on the lower endrun. The normalized \( \tau(\alpha) \) profile is compared with \( \tau_{opt}(\alpha) \). \( \tau_{opt}(\alpha) \) is a reference ideal transmission ratio which makes possible the emulation of a typical valve torque profile using a linear actuator exerting a constant force, as example an hydraulic actuator fed with constant pressure. Unfortunately the chosen transmission system meets the optimal requirements on the end-run, despite this it provide a drastic reduction of the force exerted by the actuator, especially for intermediate positions corresponding to approximately null value of the angle \( \alpha \).

![Figure 5. Normalized \( \tau(\alpha) \) compared with the reference \( \tau_{opt}(\alpha) \)](image)

Considering the specifications of TABLE I. and the geometry described in TABLE II. It’s possible to size the hydraulic cylinder which is described in TABLE III.

### TABLE III. MAIN GEOMETRIC PARAMETER OF THE CHOSEN HYDRAULIC ACTUATOR

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Int. Diameter</td>
<td>63[mm]</td>
<td>*Corresponding area of about 15 cm²</td>
</tr>
<tr>
<td>Rod diameter</td>
<td>45 mm</td>
<td></td>
</tr>
<tr>
<td>Max flow</td>
<td>4.6 liters/minute</td>
<td>Calculated for an augmented max speed of 0.04-0.05 m/s</td>
</tr>
<tr>
<td>Pressure</td>
<td>160-170 bar</td>
<td>To erogate a nominal torque of 5000Nm</td>
</tr>
</tbody>
</table>

### IV. HYDRAULIC PLANT DESIGN

The hydraulic plant of the rig is briefly described by the LMS Amesim™ scheme visible in Figure 6. The chosen plant configuration is selected in order to make possible the implementation of two different control strategies:

- Strategy “A”: Proportional Flow Controller
- Strategy “B”: Pressure Limitation with Adjustable Compliance

![Figure 6. LMS Amesim™ scheme of the rig plant](image)

**A. Proportional Flow Controller**

In this plant configuration the 4/3 distributor valve visible in Figure 7. is closed and the actuator is controlled by a proportional valve realizing the torque-force control scheme described in Figure 7. A proportional valve feed by a stabilized pressure source controls the actuator. The inlet pressure is stabilized using the servo-relief valve and the accumulator visible in the plant scheme of Figure 7.
In particular the transfer function $G_2(s)$ describes the relationship between a displacement of the cylinder and the corresponding pressure variation of the cylinder. $G_2(s)$ represents a cross coupling term between the rig actuation dynamic and the corresponding tested system, which should be cancelled or minimized to improve the dynamical response of the system.

For this reason, in the control scheme of Figure 7, is also implemented a velocity feedback contribution proportional to cylinder speed. However, dynamical performances of the system are negatively affected by both limited bandwidth of the valve and compressibility effects of oil. This approach, suggested by different sources in literature [22][25], has been yet successfully applied by Allotta and Pugi, on the force control of a railway pantographs [20]. It should be also noticed that an increase of compressibility effects (corresponding to an increase of the $V_o/E_b$ ratio) and of the leakage ($h_t$) should produce a reduction of $G_2(s)$. However, this solution has often limited benefits and potential drawbacks since the increase of the above cited terms also drastically reduce the dynamical response of the valve, corresponding to $G_f(s)$.

### B. Pressure Limitation with Adjustable Compliance

Tested positioner have a quite slow dynamical response corresponding to valve opening times of about 10-12seconds, so the high dynamical response granted by the previously described controller it’s not so important. On the other hand considering the high variability of tested components both in terms of dynamical response and simulated torques profiles, robustness, and stability performance of the controller are more important and should justify the adoption of the less conventional layout described in Figure 8.

In this case respect to the plant scheme of Figure 6, the proportional valve is deactivated. Force exerted by the actuator is controlled by regulating the pressure in one of the two chambers of the cylinder while the other one is connected to the oil tank. A 4/3 distributor is used to switch the connections of the chamber respect to the opening or closing direction of the valve.

Pressure is regulated using a servo-relief valve whose pressure reference should be continuously controlled by an external reference signal (typically an analog current reference).

As visible in the scheme of Figure 8, the reference command pressure for the relief valve, $P_{ref}$ is calculated as the sum of two contributions:

- $P_{act}$: once the desired torque profile $M_{ref}(\alpha)$ is calculated the corresponding pressure $P_{ref}(\alpha)$ that the actuator have to reproduce should be calculated according to equation (4), which is calculated considering ideal static equilibrium of the machine (no friction, neglected inertial contributions, ideal hydraulic plant response):

$$P_{act} = \frac{M_{ref}(\alpha) \cdot \frac{1}{A}}{\tau \cdot \alpha}$$

In particular the transfer function $G_2(s)$ describes the relationship between a displacement of the cylinder and the corresponding pressure variation of the cylinder. $G_2(s)$ represents a cross coupling term between the rig actuation dynamic and the corresponding tested system, which should be cancelled or minimized to improve the dynamical response of the system.

For this reason, in the control scheme of Figure 7, is also implemented a velocity feedback contribution proportional to cylinder speed. However, dynamical performances of the system are negatively affected by both limited bandwidth of the valve and compressibility effects of oil. This approach, suggested by different sources in literature [22][25], has been yet successfully applied by Allotta and Pugi, on the force control of a railway pantographs [20]. It should be also noticed that an increase of compressibility effects (corresponding to an increase of the $V_o/E_b$ ratio) and of the leakage ($h_t$) should produce a reduction of $G_2(s)$. However, this solution has often limited benefits and potential drawbacks since the increase of the above cited terms also drastically reduce the dynamical response of the valve, corresponding to $G_f(s)$.

### B. Pressure Limitation with Adjustable Compliance

Tested positioner have a quite slow dynamical response corresponding to valve opening times of about 10-12seconds, so the high dynamical response granted by the previously described controller it’s not so important. On the other hand considering the high variability of tested components both in terms of dynamical response and simulated torques profiles, robustness, and stability performance of the controller are more important and should justify the adoption of the less conventional layout described in Figure 8.

In this case respect to the plant scheme of Figure 6, the proportional valve is deactivated. Force exerted by the actuator is controlled by regulating the pressure in one of the two chambers of the cylinder while the other one is connected to the oil tank. A 4/3 distributor is used to switch the connections of the chamber respect to the opening or closing direction of the valve.

Pressure is regulated using a servo-relief valve whose pressure reference should be continuously controlled by an external reference signal (typically an analog current reference).

As visible in the scheme of Figure 8, the reference command pressure for the relief valve, $P_{ref}$ is calculated as the sum of two contributions:

- $P_{act}$: once the desired torque profile $M_{ref}(\alpha)$ is calculated the corresponding pressure $P_{ref}(\alpha)$ that the actuator have to reproduce should be calculated according to equation (4), which is calculated considering ideal static equilibrium of the machine (no friction, neglected inertial contributions, ideal hydraulic plant response):

$$P_{act} = \frac{M_{ref}(\alpha) \cdot \frac{1}{A}}{\tau \cdot \alpha}$$

In particular the transfer function $G_2(s)$ describes the relationship between a displacement of the cylinder and the corresponding pressure variation of the cylinder. $G_2(s)$ represents a cross coupling term between the rig actuation dynamic and the corresponding tested system, which should be cancelled or minimized to improve the dynamical response of the system.

For this reason, in the control scheme of Figure 7, is also implemented a velocity feedback contribution proportional to cylinder speed. However, dynamical performances of the system are negatively affected by both limited bandwidth of the valve and compressibility effects of oil. This approach, suggested by different sources in literature [22][25], has been yet successfully applied by Allotta and Pugi, on the force control of a railway pantographs [20]. It should be also noticed that an increase of compressibility effects (corresponding to an increase of the $V_o/E_b$ ratio) and of the leakage ($h_t$) should produce a reduction of $G_2(s)$. However, this solution has often limited benefits and potential drawbacks since the increase of the above cited terms also drastically reduce the dynamical response of the valve, corresponding to $G_f(s)$.

### B. Pressure Limitation with Adjustable Compliance

Tested positioner have a quite slow dynamical response corresponding to valve opening times of about 10-12seconds, so the high dynamical response granted by the previously described controller it’s not so important. On the other hand considering the high variability of tested components both in terms of dynamical response and simulated torques profiles, robustness, and stability performance of the controller are more important and should justify the adoption of the less conventional layout described in Figure 8.

In this case respect to the plant scheme of Figure 6, the proportional valve is deactivated. Force exerted by the actuator is controlled by regulating the pressure in one of the two chambers of the cylinder while the other one is connected to the oil tank. A 4/3 distributor is used to switch the connections of the chamber respect to the opening or closing direction of the valve.

Pressure is regulated using a servo-relief valve whose pressure reference should be continuously controlled by an external reference signal (typically an analog current reference). As visible in the scheme of Figure 8, the reference command pressure for the relief valve, $P_{ref}$ is calculated as the sum of two contributions:

- $P_{act}$: once the desired torque profile $M_{ref}(\alpha)$ is calculated the corresponding pressure $P_{ref}(\alpha)$ that the actuator have to reproduce should be calculated according to equation (4), which is calculated considering ideal static equilibrium of the machine (no friction, neglected inertial contributions, ideal hydraulic plant response):

$$P_{act} = \frac{M_{ref}(\alpha) \cdot \frac{1}{A}}{\tau \cdot \alpha}$$
In particular considering the plant layout of Figure 6, the dynamical behaviour of the system is described by (5) in which the following simplifications are assumed to be valid:

- The pumping unit assures a constant flow \( Q_{\text{pump}} \).
- The flow delivered to the controlled cylinder chamber \( Q \).
- The servo-relief valve flow \( Q_{\text{valve}} \) is constrained to be negative since the valve should be used only to discharge the controlled capacities. For the sake of simplicity the flow of the valve is supposed to be controlled by a proportional regulator with a constant gain \( k \). Valve dynamics is reproduced as a second order system with an Eigen frequency \( \omega_n \) of about 12 Hz and a damping coefficient of about 0.7,0.8.

\[
Q + Q_{\text{pump}} + Q_{\text{valve}} = k(P_{\text{ref}} - P) \frac{\alpha_s^2}{\omega_n^2 + 2\omega_n\xi + \omega_n^2} + Q_{\text{pump}} + \left( -\frac{A_x s + \frac{V_r}{V_e} s P}{E_{\text{ref}}} \right) = 0
\]  

(5)

In (5) the equivalent compressibility term \((V_{\text{ref}}E_{\text{ref}})\) is highly influenced by the presence of the air or nitrogen filled accumulator. In particular the compressibility of the pneumatic accumulator is largely dominant and should be approximately calculated according (6)

\[
\frac{dV_{\text{ref}}}{dP} \approx \frac{V_{\text{ref}}}{k(P_{\text{ref}}V_{\text{ref}})}
\]  

(6)

Where the following symbology is adopted:

- \( V_{\text{ref}} \) is the current volume of the pneumatic accumulator, while \( P_{\text{ref}} \) and \( V_{\text{ref}} \) represent the initial (preload) pressure and volume values.
- \( \beta \) is the coefficient of the polytropic transformation which is used to approximate the behavior of the gas in the accumulator.

Solving (5), it’s possible to calculate the behavior of the pressure \( P \) of the controlled chamber of the actuator as a function of three inputs, \( Q_{\text{pump}}, P_{\text{ref}}, \alpha \).

\[
P = \frac{Q_{\text{pump}}}{k} \left( \frac{1}{\omega_n^2 + 2\omega_n\xi + \omega_n^2} \right) + \frac{P_{\text{ref}}}{1 + \frac{V_{\text{ref}}}{kE_{\text{ref}}\alpha_s^2} \left( 1 + 2\alpha_s + \frac{V_{\text{ref}}}{kE_{\text{ref}}} \right)} + \frac{\alpha_s^2}{\omega_n^2 + 2\omega_n\xi + \omega_n^2} + \frac{\alpha_s^2}{1 + \frac{V_{\text{ref}}}{kE_{\text{ref}}\alpha_s^2} \left( 1 + 2\alpha_s + \frac{V_{\text{ref}}}{kE_{\text{ref}}} \right)}
\]  

(7)

Since the area of the actuator \( A \) and the transmission ratio \( \pi(\alpha) \) are known, it’s possible to calculate from (7) the three transfer functions \( G_{\text{pump}}(s), G_{\text{ref}}(s), K_{\text{ref}}(s) \), between the three inputs \( Q_{\text{pump}}, P_{\text{ref}}, \alpha \) and the torque \( M \) delivered by the actuator (8).
\[ M = Q_{\text{pump}}G_{\text{pump}}(s) + G_{\text{ref}}(s) + xK_{\text{rip}}(s) \]  
\[ G_{\text{pump}}(s) = \frac{\tau(\alpha)A}{k \left( \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} + \frac{V_{op}}{kE_{op}} \right)} \]  
\[ G_{\text{ref}}(s) = \frac{\tau(\alpha)A_2}{\left( 1 + \frac{V_{op}}{kE_{op}\omega_n^2} s^2 + 2\xi\omega_n s + \omega_n^2 \right)} \]  
\[ K_{\text{rip}}(s) = \frac{-\tau(\alpha)A^2 s}{k \left( \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} + \frac{V_{op}}{kE_{op}} \right)} \]

Some further considerations should be performed from (8):

- **\( G_{\text{pump}}(s) \):** \( Q_{\text{pump}} \) introduces a disturbance whose influence is rejected by the pressure control loop of the relief valve. In fact, higher valve gain \( k \) decreases the module of \( G_{\text{pump}}(s) \).
- **\( G_{\text{ref}}(s) \):** it’s possible to calculate a root locus of \( G_{\text{ref}}(s) \) parametrized respect to valve gain \( k \) as visible in Figure 10. For too high values of \( k \) the system became clearly unstable.

**K_{\text{rip}}(s):** this transfer function represent the coupling term with the tested system. As a consequence to increase the robustness without penalizing to much the gain \( k \), \( K_{\text{rip}}(s) \) has to be carefully shaped. For position disturbances with frequencies far higher than the frequency of the valve \( \omega_n \), \( K_{\text{rip}}(s) \) is approximated by (9).

\[ s \gg \omega_n \Rightarrow K_{\text{rip}}(s) \approx \frac{-\tau(\alpha)A^2 E_{op}}{V_{op}} \]  

From (9) it should be deduced that by moderately increasing the compressibility term \( V_{op}/E_{op} \), it’s possible to reduce the value of \( K_{\text{rip}}(s) \) without compromising to much the bandwidth of the pressure controller which should be calculated from the expression of \( G_{\text{ref}}(s) \).

### V. PRELIMINARY SIMULATION USING LMS AMESIM

#### A. Model Description

Using LMS Amesim™, authors developed a complete model of the rig whose main feature are visible in Figure 11.

In particular both rig and tested actuator mechanical behavior are reproduced in complete multibody model. The geometry of the rig is the same described in TABLE II. The tested actuator and positioner is supposed to be a scotch yoke transmission with the geometry described in Figure 1. Inertial properties of components are taken directly from CAD geometries. Tested actuator is supposed to be fed with an air pressure corresponding to a maximum opening torque of about 5000Nm. The controller of the positioner is supposed to be an high gain PI (Proportional Integrator) regulator manually tuned with an equivalent impedance of the valve in order to obtain a feasible response. For the rig controller both the two regulator configurations are compared.

The response of hydraulic [26] and pneumatic components in terms of hydraulic losses and dynamical behavior is modelled including pipes which are modelled considering RI-C [28],[29], components in which resistive, inertial and capacitive effects working on fluid inside the pipe are evaluated.

Other important non linearities as friction and compliances on joints of mechanical components are modeled in order to further verify the robustness of the proposed controller. In particular on prismatic joints, kinematic and a static friction factors are evaluated to be equal respectively to 0.15 and 0.3.

Also on rotoidal joints friction is applied considering the application of the same friction model [30] over a cylindrical surface with a radius of about 10mm.

![Figure 10. Root Locus of G_{ref}(s)](image)

![Figure 11. LMS Amesim model of the HIL Test rig](image)

Finally mechanical compliances on joints are introduced as equivalent springs/bushings which reproduce a limited deformation of the joint supposed to be linear respect to the applied load.
B. Preliminary Simulation Results

Using the model described above, authors compared the performances of both the proposed controllers in terms of robustness. Both proportional flow controller, and the pressure limitation one have been successfully calibrated for a nominal condition in which both the controller are able to produce an almost equivalent performance. Then a Montecarlo simulation with perturbed parameters is performed: the model is perturbed considering some parametric variations, as example in the simulated friction factor, or in the reference torque profile that have to be simulated, and the effects in terms of robustness of response of the system are evaluated. Some qualitative results are summarized in TABLE IV. and in TABLE V.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Perturbation(s) State</th>
<th>Flow Controller</th>
<th>Pressure Limitation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction on joints</td>
<td>No-Friction/ Doubled Friction</td>
<td>Robust</td>
<td>Robust</td>
</tr>
<tr>
<td>Compliance on Joints</td>
<td>No Compliance/ 10X compliance</td>
<td>Sensitive</td>
<td>Robust</td>
</tr>
<tr>
<td>Scaled Ref. Torque Profile</td>
<td>Max. Torque Profile/ Min. Torque Profile</td>
<td>Very sensitive</td>
<td>Sensitive</td>
</tr>
<tr>
<td>Variation of the positioner loop Gains</td>
<td>Pos. Gain mult. x 0.5/ Pos. Gain mult. x 10</td>
<td>Very sensitive</td>
<td>Robust</td>
</tr>
</tbody>
</table>

Preliminary simulations substantially confirm the behaviour predicted by the theoretical analysis performed on linearized models of the system: in particular, the second strategy seems to be quite more robust respect to most of the uncertainties due to parametric variations of the tested system. In particular, the uncertainties of the simulated positioner were expressed in terms of gains of its position loop and in terms of exerted torque: a variation of the gain of the position loop of the positioner was used to roughly reproduce the behaviour of a system with different stability margins and performances. On the other hand, the simulations of pneumatic actuators, scaled to cover different torque categories, involved different system sizing and consequent variations of the equivalent mechanical impedance of the actuator virtually tested on the rig. In particular in Figure 12, some results concerning the simulation of different $M_{ref}$ profiles and the corresponding $M$ measured for different friction factor are shown: it’s clearly noticeable that performances and robustness of the controller are more influenced by a variation of the simulated torque profile respect to even big variation of the simulated friction on joints. As visible in Figure 13, the contribution in terms of actuator pressure needed to compensate the friction is quite small, explaining the reason of the relative robustness of the control loop respect to friction variations.

VI. CONCLUSIONS AND FUTURE DEVELOPMENTS

In this work the preliminary study of a Hardware in the loop rig for the testing of quarter valve actuation and position control system has been presented. In particular both actuation and control system have been optimized in order to increase robustness and performance respect to quite demanding specifications especially in terms of robustness respect to parametric uncertainties of the rig and variability of the tested system. The major contribution of this work is the fruitful
transfer and application of know-how which is more frequently used in robotics, to an industrial testing application. For the next year the rig will be effectively put in service and authors hope to be able to discuss performance and robustness of the tested system with the support of fresh experimental data.

VII. ACKNOWLEDGEMENTS

This work was financed as a part of the project High Efficiency Valves (CUP: D55C12009530007) financed by the program POR CRO FESR of Regione Toscana (European Funds for Regional Industrial R&D projects). Apart funding, which is a fundamental contribution for research, authors wish also to tanks engineering students of university of Florence for their contribution and for their curious and enthusiastic approach and in particular Giuseppe Occhipinti and Andrea Pulcinelli. In addition authors wish to gratefully remember all the people of Velan ABV SPA which have contributed to this work with their experience and competence, in particular Fabio Lapini and Michele Graziani.

VIII. REFERENCES