ANALYSIS OF THE BEHAVIOUR OF A WATER HYDRAULIC CRANE

Simone Bruno[1], Harri Sairiala[2] and Kari T. Koskinen[3]

[1]Politecnico di Bari, DIMeG, Via Re David 200, 70122 Bari, Italy
s.bruno@poliba.it
[2]Tampere University of Technology, Institute of Hydraulic and Automation, Valkeakoski Research Unit, Valkeakoski, Finland
harri.sairiala@tut.fi
[3]Tampere University of Technology, Institute of Hydraulic and Automation, Tampere, Finland
kari.t.koskinen@tut.fi

Abstract

In this work a hydraulic crane was equipped with water hydraulic components. A control system using two PID controllers plus a Feed Forward block (F-PID) was implemented and in order to study the dynamic behaviour of this kind of system a series of tests were performed. At first, a characterisation of the system (mechanical and hydraulic) was performed and then the different steps of the controllers design were analysed, by means of various end-effector trajectories. The results show a very good behaviour of this kind of system with a low positioning error.

Keywords: hydraulic crane, water hydraulic, PID, proportional valve

1 Introduction

In the ending of 20th century, many factors contributed to a new development of the water hydraulic (Trostmann 1996). In particular, a great attention to the environmental problem and the health of the product consumers focused the attention on water as pressure medium instead of the mineral oil. Moreover, an improvement of the technologies (new material with high corrosion resistance, high precision in the machining that allows very small tolerance, etc.) allows obtaining more competitive water hydraulic components. Indeed, the use of water as pressure medium instead of mineral oil has different advantages. There are many application areas for water hydraulics to substitute traditional oil hydraulics or, in particular fields, electric motors and pneumatic actuators (Krutz et al., 2004, Lim et al., 2003, Rydberg 2001).

Many work, recent or not, on the control of hydraulic system by means of different techniques (Andersen et al. 2005, Kim et al. 2005, Saarela et al. 2005) exist, and most parts of these use oil as a fluid medium. In this study an analysis of the dynamic behaviour of a Water Hydraulic crane will be presented. In the last years many Research Centres have been interested in the Water Hydraulic Technology and many works in this field have been realised and presented. Usually, in the most important conferences in Fluid Power a section of Water Hydraulic works is present.

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In this work a hydraulic crane of the HIAB Company (HIAB 031) was equipped with water hydraulic components (cylinders and proportional valves) and a control system was realised in order to evaluate the positioning capability of this kind of system. Many works on the oil hydraulic systems exist in Literature and different kinds of controllers have been elaborated and implemented (Andersen et al., 2005, Bonchis et al., 2001, Kim et al., 2005, Sohl et al., 1999).

To realise this work, different steps were necessary. The first step was a characterisation of the system in order to design the controllers. A series of tests were performed in order to know the instability limits of the crane system. In this first phase of experimental tests the crane was unloaded.

In the second phase, after the design and the testing of the controllers, a load was applied in order to know the system behaviour in a working condition.

As said before, the most part of the papers in the control of a hydraulic system are in oil hydraulic (Andersen et al., 2005, Budny et al., 2003, Ha et al., 2000, Sohl et al. 1999). For this reason in this work, a PID-controller tuned by means of the Ziegler-Nichols method was preferred to a more complex controller, in order to understand the real capability of this system equipped with LPWH (Low Pressure Water Hydraulic) components (about 50 bars). It is possible that a more complex control technique allows to improve the performance and reduce the positioning error but this is a next step. The first step is the understanding of the capability of such a system. Then the results of the experimental tests have been analysed.

Fig. 1: *The HIAB hydraulic crane*

2 Crane System

The design of the controller needed a mechanical and hydraulic characterisation of the whole system.

The crane is a HIAB 031 equipped with LPWH (Low Pressure Water Hydraulic) components. The original oil hydraulic cylinders were substituted by water hydraulic cylinders, but the kinematics of the system did not change. In this work, only the lift and the tilt actuator where installed and each cylinder was controlled by means of a water hydraulic proportional valve (Takahashi et al., 1999).

In Fig. 2 the hydraulic system is shown. By means of a relief valve and an accumulator, the pressure supply is kept at the constant value of 50 bars. From the supply, the flow is sent to the cylinder through two proportional valves.

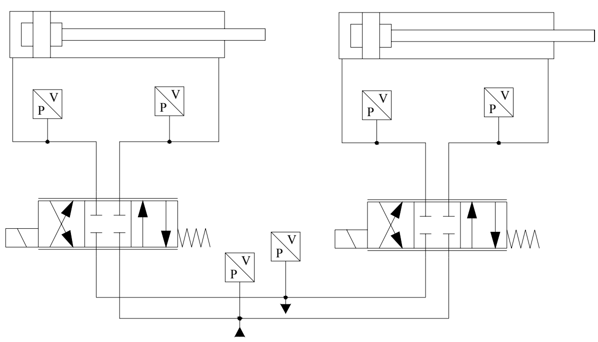

Fig. 2: *Hydraulic system*

To acquire the data from the system, there were:

6 pressure transducers, 2 for each actuator (1 on the piston side and 1 on the rod side), one for the tank pressure and one for the supply pressure:

- 2 pulse encoders for the cylinder displacements;
- 2 LVDT integrated on the proportional valves.

The data acquisition and the controlling of the system were realised by means of a DSpace® system coupled to a Simulink® code.

2.1 Kinematics of the Crane

The crane is a two degrees of freedom system, moving in the vertical plane (Beiner, 1997) as shown in Fig. 3. The configuration of the crane can be indicated by means of the two joint angles θ_1 and θ_2.

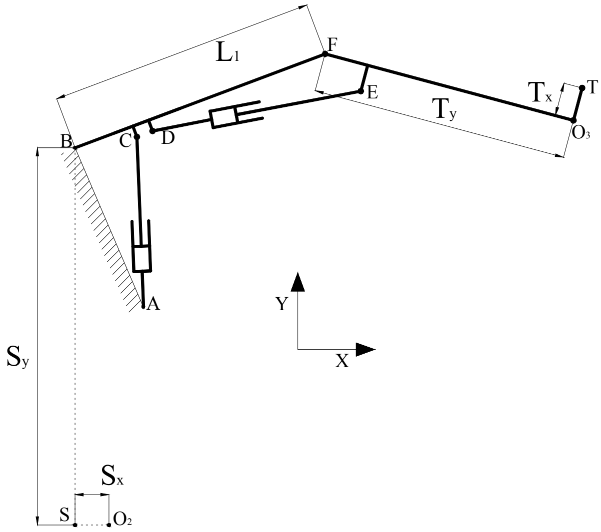

Fig. 3: *Kinematics scheme of the crane*

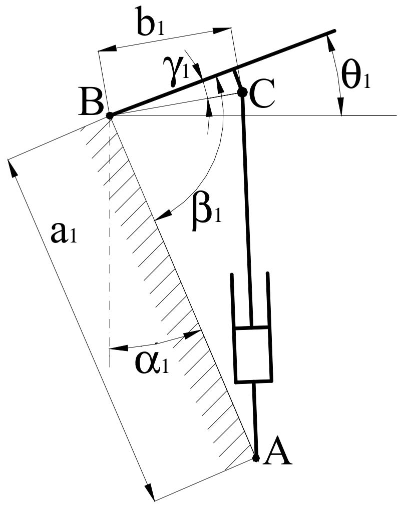

Fig. 4: *Kinematics scheme of the 1st frame*

The relationship between the joint angles and the cylinder displacements can be obtained by means of geometrical considerations (Fig. 4 and 5).

Considering the lift cylinder displacement x_1, the value of joint angle of the first frame is given by the following equation:

$$\theta_1 = \alpha_1 + \arccos\left[\frac{a_1^2 + b_1^2 - (l_1 + x_1)^2}{2a_1 b_1}\right] + \gamma_1 - 90° \quad (1)$$

where the parameter l_1 is the total length of the cylinder when the piston displacement is null.

The relationship between the joint angle of the second frame and the displacement of the tilt actuator can be expressed in a way similar to the previous:

$$\theta_2 = \alpha_2 + \arccos\left[\frac{a_2^2 + b_2^2 - (l_2 + x_2)^2}{2a_2 b_2}\right] + \gamma_2 - 180° \quad (2)$$

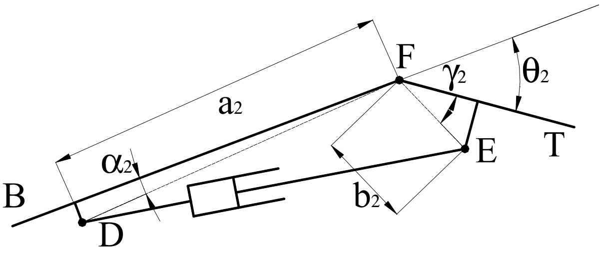

Fig. 5: *Kinematics scheme of the 2nd frame*

Finally, the crane end-effector position in the vertical plane XY, indicated by the point *T* in Fig. 3, can be expressed by means of the joint angle values:

$$\begin{cases} x = \cos(\theta_1 + \theta_2)T_y - \sin(\theta_1 + \theta_2)T_x + L_1 \cos\theta_1 - S_x \\ y = \sin(\theta_1 + \theta_2)T_y - \cos(\theta_1 + \theta_2)T_x + L_1 \sin\theta_1 + S_y \end{cases} \quad (3)$$

The workspace of the crane is shown in Fig. 6.

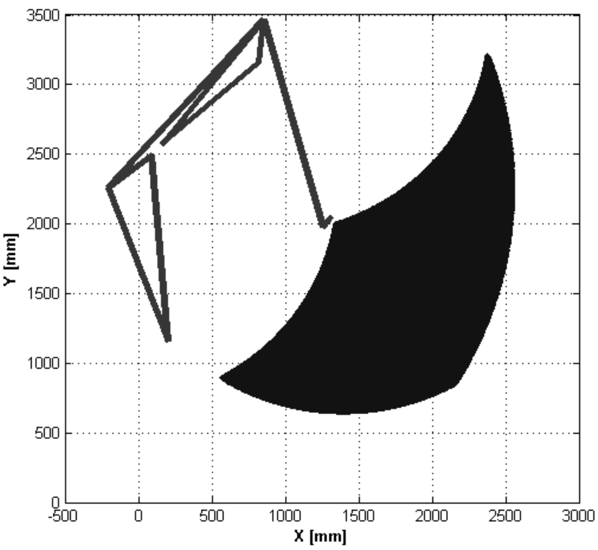

Fig. 6: *Workspace of the crane*

2.2 Dynamic Characterisation of the Crane

In order to design the controllers of the system a series of tests were performed. These tests allow knowing the instability conditions of the crane in the different configurations. A simple P-controller was used for each actuator and the value of the P-gain was increased until the crane reached instability by a step command. The values of the P-gains and of the oscillation periods were used to design the PID controllers by means of the Ziegler-Nichols method. Although Ziegler-Nichols method is not very sophisticated, it is a simple method and it is sufficient in order to realise the purpose of the present work.

The cylinder can be considered as a 2nd order system, where the mass is the mass of the piston and the load, and the spring is due to the elastic behaviour of the fluid in the two cylinder chambers and the pipes connected to it (Budny et al., 2001, Koivisto et al., 2005, Krus 1988).

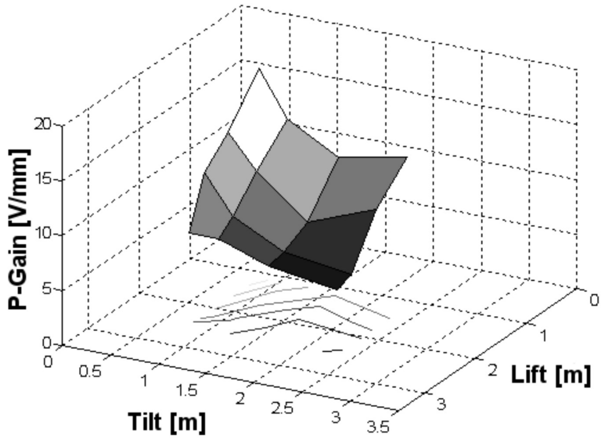

Fig. 7: *Critical lift P-gain*

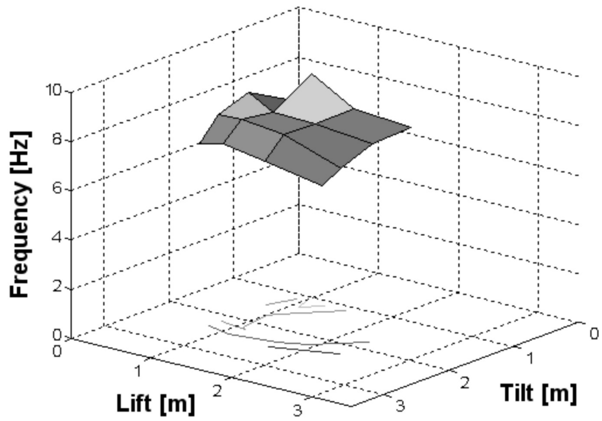

Fig. 8: *Critical lift oscillation frequencies*

Table 1: Critical tilt P-gains and oscillations periods

Stroke [%]	Critical P-gain [V/m]	Frequency [Hz]
20	12800	4.9
40	14000	4.6
60	12400	4.7
80	12400	5

For this reason in order to obtain the critical P-gains, different configurations were used. For the designing of the tilt controller four different configurations of the tilt cylinder were used. In fact, considering the tilt frame, the inertial characteristic is always the same, while the elastic characteristic changes with the piston position. The critical P-gains and the oscillation periods for the tilt controller are shown in the Table 1.

In Fig. 7 and 8, the critical P-gains and the oscillation periods for the lift are shown. For the lift frame, in order to study its behaviour in the different part of the workspace, 16 different configurations were used, 4 configurations for the lift and for each of these there were 4 different tilt displacements.

3 Experimental Tests

Each actuator was controlled by means of an F-PID, like that shown in Fig. 9. Often, in the position control, the Feed Forward is proportional to the velocity reference (Virvalo et al., 2000). In this work the usage of a static model of the proportional valve was preferred similarly to other works (Qui et al., 2003, Saarela et al., 2005). The flow rate necessary to obtain that cylinder velocity is computed from the velocity reference:

$$v_{\text{ref}} = \frac{dx_{\text{ref}}}{dt} \Rightarrow Q_{\text{ref}} = A_c v_{\text{ref}} \tag{4}$$

The valve opening necessary to obtain that flow rate is calculated then from the pressure:

$$Q_{\text{ref}} = A_v \sqrt{\frac{2\Delta p}{\rho}} \Rightarrow y_v = f(A_v) = \left(Q_{ref} \sqrt{\frac{\rho}{2\Delta p}} \right) \tag{5}$$

Finally, the voltage signal sent to the proportional valve is calculated from the desired spool position (Takahashi et al., 1999, Sairiala et al., 2004):

$$u = u(y_v) \tag{6}$$

where $u=u(y_v)$ is the relationship between the static position of the spool y_v and the voltage signal sent to the valve u.

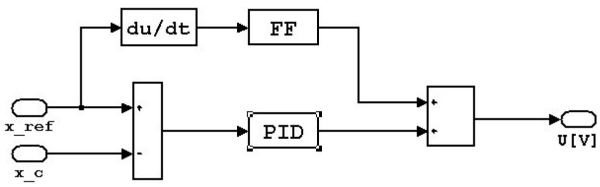

Fig. 9: *F-PID controller*

3.1 Tests without Load

The first series of test were performed without load on the crane end-effector.

Usually the hydraulic cylinder is a system that needs an integrative-gain in order to improve the steady state accuracy. For this reason, in a steady positioning test it is possible to notice at beginning a great overshoot and then the limit cycles around the desired position (Virvalo, 2001). In order to reduce these two problems, two simple solutions are introduced in the PID controller.

The first solution, to reduce the overshoot, is the introduction of a saturation limit for the integrative signal. In fact, even if the voltage value sent to the proportional valve is limited in the range [-10 V, 10 V], the integral of the error can become very high and it needs a great overshoot to be compensated. The introduction of the saturation limit in the integrative component of the PID controller fixes a maximum (or minimum, if it is negative) value of the integral and consequently it fixes a limit for the overshoot.

The second solution, in order to reduce the limit-cycle oscillations, is the introduction of a switch for the integrative component. When the error is in a desired range (for instance in the range [-0.1 mm, 0.1 mm]) the integrative component is switched off. In this way in that range the integrative-block does not act and there are not these oscillations. The integral block switch (ISR) has some advantages and some disadvantages. The most important disadvantage is that the static error cannot be equal to zero, but equal at the minimum value between the error due to proportional-gain and the switch range. The most important advantage is that the limit-cycle oscillations are reduced. So, in order to decide the optimal value of the switch range, an evaluation of the advantages and disadvantages is necessary.

Three different series of tests was performed:

- A step command with and without path controlling;
- A circular path with different centres, radius and tangential velocities;
- A complex path, constituted by four points connected by straight line.

In all the series of tests, the value of the gains are equal (see Table 2), while in every series of tests different switch ranges are used. It must be noted that the Ziegler-Nichols method was considered to design the controllers but then in the experimental tests gains farther from the critical conditions are used.

Table 2: Gains of the PID controllers

Actuator	P-gain [V/m]	I-gain [V/ms]	D-gain [Vs/m]
Lift	2000	20000	40
Tilt	4000	40000	160

In the step command test the end-effector is moved from the point P1(1.30, 1.00) to the point P2(2.15, 1.80) and then from P2 to P1, without path-control. In Fig. 10 and 11 the results without the usage of the integral switch range are shown.

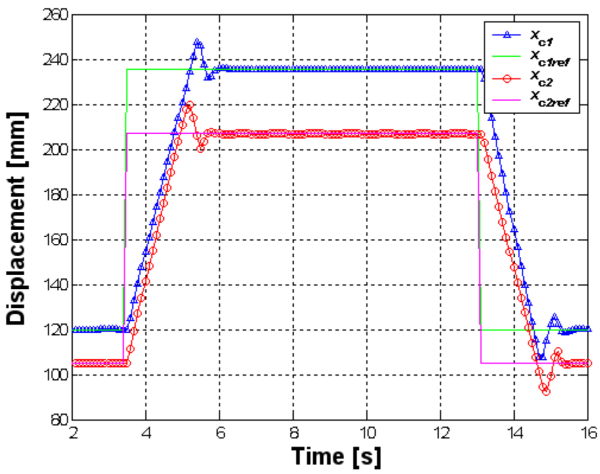

Fig. 10: *Cylinder displacements (without ISR)*

The Fig. 10 shows the cylinder displacements. In this diagram it is possible to notice the overshoots and then the steady state conditions. Moreover in this diagram, where the whole displacements are shown, it is possible to notice the limit cycles due to the integrative components of the controllers. This effect is better shown in Fig. 11, where there is a zoom of the cylinder errors after the overshoots. The cylinders reach the desired positions, at first there are the overshoots due to the integrative components of the PID controllers and then there are the limit-cycle oscillations.

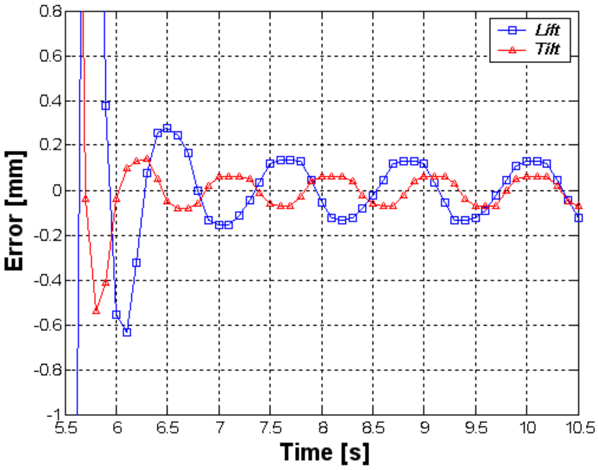

Fig. 11: *Zoom of the displacement errors (without ISR)*

After the step movements of the cylinders, there are oscillations around the steady and desired displacements. The amplitude of these oscillations is less than 0.2 mm for the lift and less than 0.1 mm for the tilt. This means these amplitude values can be used to set the values of the integral switch ranges.

The results of the step command with the I.S.R set to 0.2 mm for the lift and 0.1 mm for the tilt are shown in Fig. 12 and 13. In Fig. 12, the cylinder displacements are shown. The behaviour of the system is very similar to the previous test but the limit-cycle oscillations are not visible. In fact, better shown in Fig. 13, the diagram of the errors obtained by using the ISRs shows that these oscillations are reduced.

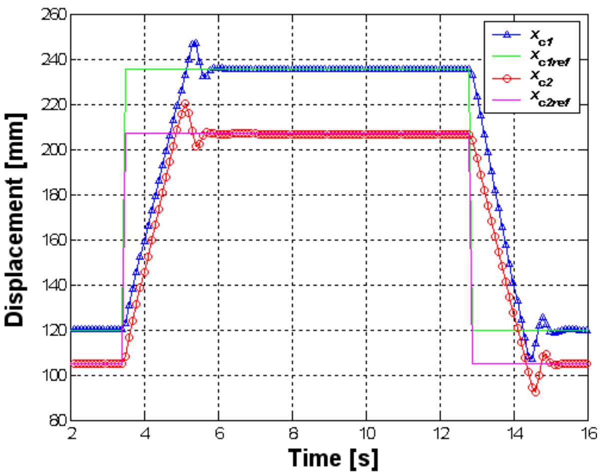

Fig. 12: *Cylinder displacements (with ISR)*

In the previous tests, the presence of the constant slope of the lift and tilt motion curves means that the flow is saturated until the target positions are reached. In order to avoid this saturation and to obtain a more smooth motion a series of tests with the same controller parameters, the same trajectory and a path control by a 5[th] order polynomial interpolation has been performed. In Fig. 14 and 15 the results are shown.

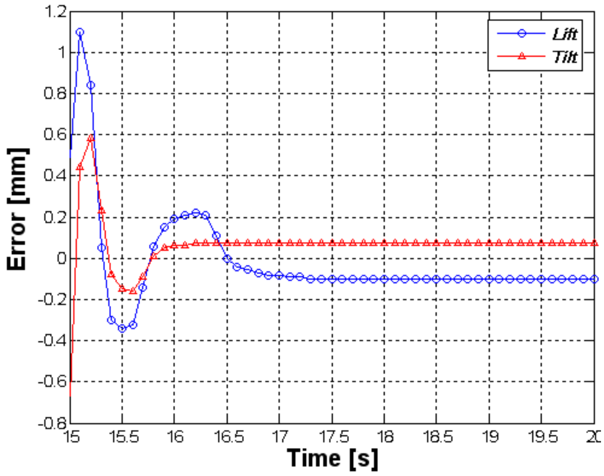

Fig. 13: *Zoom of the displacement errors (with ISR)*

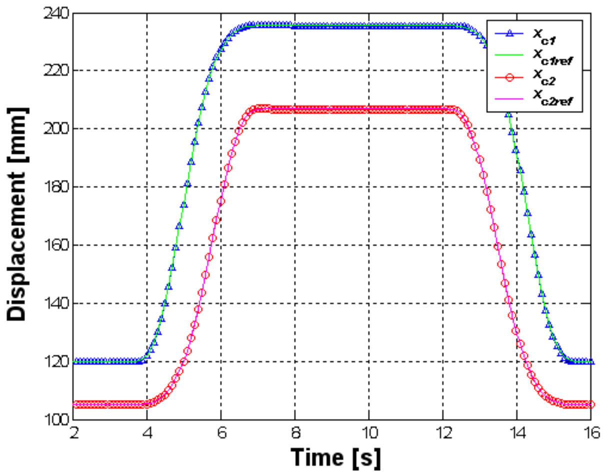

Fig. 14: *Cylinder displacements (with ISR)*

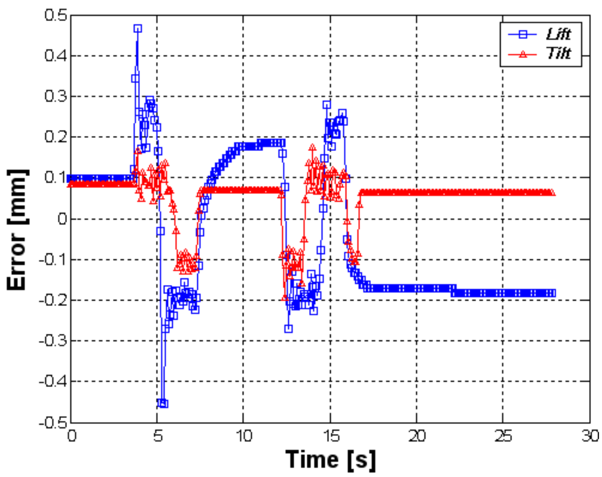

Fig. 15: *Displacement errors (with ISR)*

Figure 14 shows the cylinder displacements for this trajectory. These oscillations are better shown in Fig. 15, in the displacement errors diagram. The amplitude values of the oscillations are less than 0.2 mm for the lift cylinder and less than 0.1 mm for the tilt as obtained in the step command without path control. The error diagram shows also that the maximum errors are about 0.7 mm for the lift and 0.2 mm for the tilt.

In Fig. 15 the displacement errors are shown. The error values are very low. They are less than 0.3 mm for the lift, except at the beginning of the acceleration and deceleration phases and at maximum equal to 0.1 mm for the tilt cylinder.

3.2 Tests with the Load

In the second phase of testing and design of the controllers, a series of tests with a load on the crane end-effector ha been performed. A load of 100 kg was joined to the crane as shown in the picture of the Fig. 16.

In order to evaluate the dynamic behaviour of the crane, a step command, a circular path and a 4 points trajectory has been used.

In the following diagrams the result obtained by using a circular path are shown. The circle has the centre in O(1.85,1.4), a radius of 0.5 m and a tangential velocity equal to 0.25 m/s.

The parameters of the two controllers were:

- K_P=1600 V/m for the lift and K_P=3200 V/m for the tilt;
- K_I=16000 V/ms for the lift and K_I=32000 V/ms;
- K_D=32 Vs/m for the lift and K_D=128 Vs/m for the tilt;
- ISR=0.2 mm for the lift and ISR=0.1 mm for the tilt.

Fig. 16: *Load on the crane end-effector (100 [kg])*

Figure 17 shows the cylinder displacements for this test. The circular path in the workspace, by means of the geometrical relationship between the end-effector position and the cylinder displacement, is transformed in a "quasi" sinusoidal trajectory for each actuator.

Figure 18 shows the displacement errors. This diagram shows that the errors are equal or less than 0.2 mm for the lift and equal or less than 0.1 mm for the tilt, during the most part of the motion. There are some peaks, with a very short time duration, and with a maximum value not very high (about 0.6 [mm] for the lift and 0.35 [mm] for the tilt).

These peaks appear when the cylinders change direction and, for a short time, they are stopped (Fig. 20 and 21). This is due essentially to two reasons. The first reason is that in this kind of components, without oil lubrification, there is a high value of the friction in particular of the static friction. Different kinds of the friction force model exist and the most common is the Stribeck model (Alleyne et al., 2000, Bonchis et al., 2001, Rahmfeld et al., 2001) shown in Fig. 19. So when the cylinder changes direction, there is a time when its velocity is very low. The friction is the static friction with a high value and the piston is stuck. The second reason is that the proportional valves used in this system have a great overlap, so when the cylinders change direction, there is a quite long part of the spool stroke that does not allow the flow through the valve and consequently the cylinders cannot move.

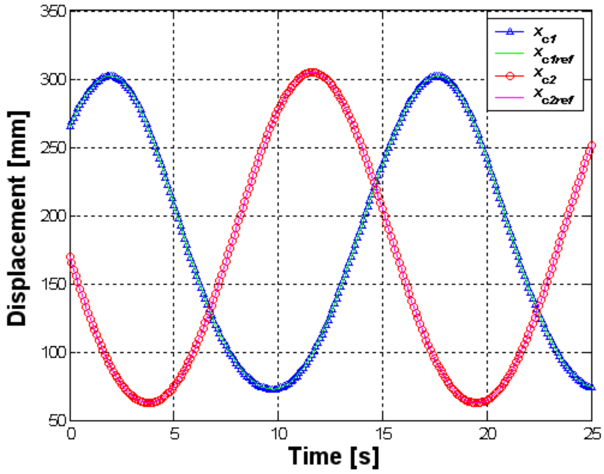

Fig. 17: *Cylinder displacements*

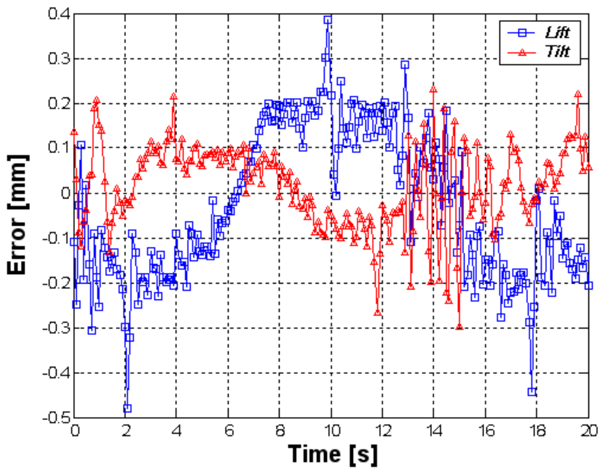

Fig. 18: *Displacement errors*

It is reasonable to think it is possible to reduce this error by using two possible solutions:

- A friction compensator. There is a need of a experimental evaluation of the different parameters of the friction force in order to realise a friction force model very close to the actual behaviour;
- A dead-band compensation. It is necessary to increase the spool velocity when it is in the centre position. In this way it is possible to reduce the time length of the no-flow condition.

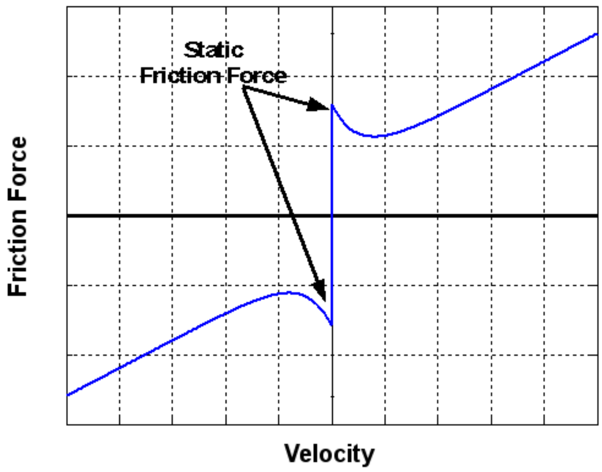

Fig.19: *Qualitative behaviour of the friction force*

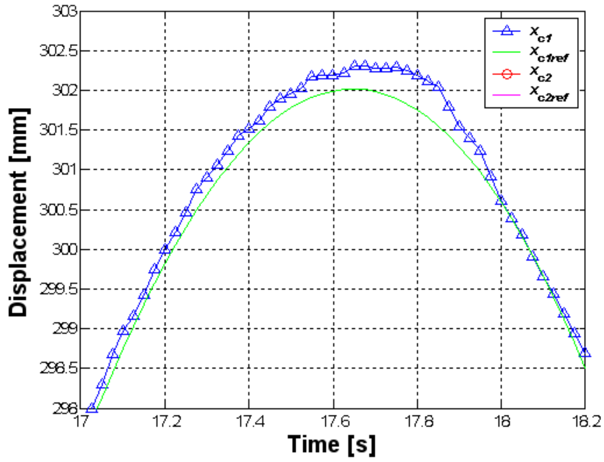

Fig. 20: *Zoom of the lift displacement*

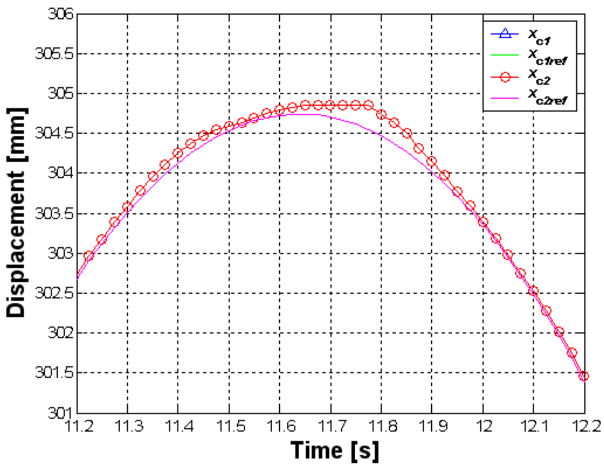

Fig. 21: *Zoom of the tilt displacement*

In Fig. 23 and 24 the comparison between a circular test with load and without load is shown.

The parameters used in this test are:

- Radius $R = 0.2$ m;
- Tangential velocity $v = 0.2$ m/s;
- Circle centre O(1.5, 1.5);

The same controller parameters were used in order to better compare the two tests. In particular, the parameters of the two controllers were:

- K_P=2000 V/m for the lift and K_P=4000 V/m for the tilt;
- K_I=20000 V/ms for the lift and K_I=40000 V/ms;
- K_D=40 Vs/m for the lift and K_D=160 Vs/m for the tilt;

The Integral Switch Ranges were not used in this test.

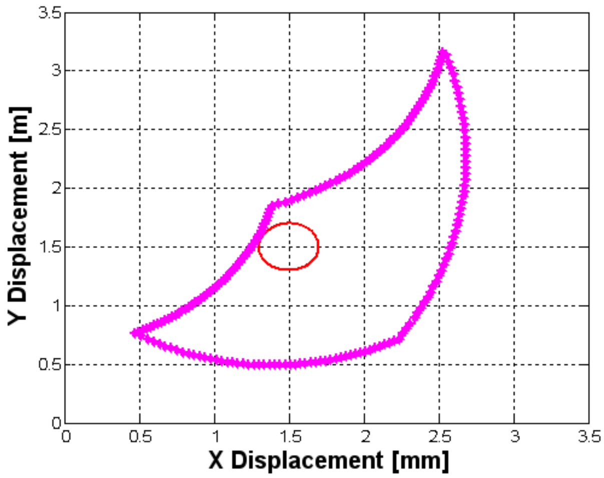

Fig. 22: *Trajectory in the workspace*

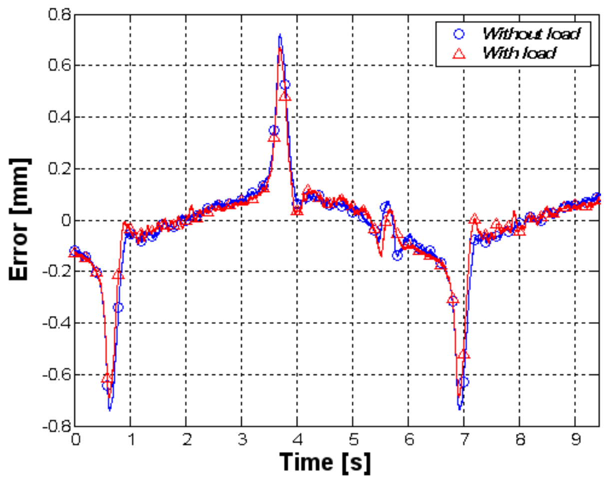

Fig. 23: *Lift displacement*

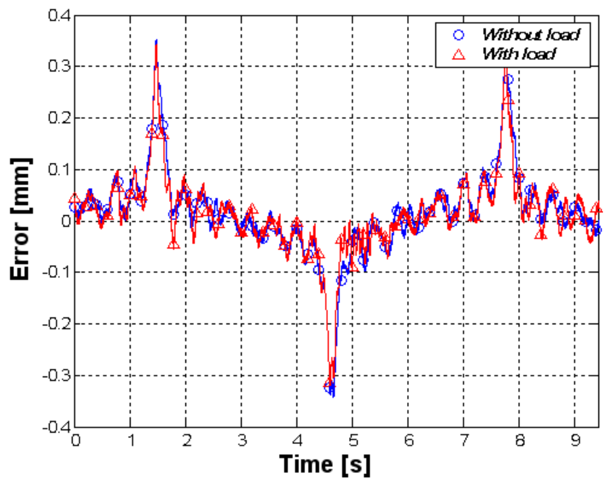

Fig. 24: *Tilt displacement*

The Fig. 23 and 24 show the lift and the tilt displacement errors respectively. In these two diagrams it is possible to notice that the behaviour of the crane does not have relevant behaviour change.

After these circular paths, a more complex trajectory composed by four points connected my means of a straight line was used. This trajectory (Fig. 25) was used to move the crane end-effector and to notice the system behaviour in the different position in the workspace.

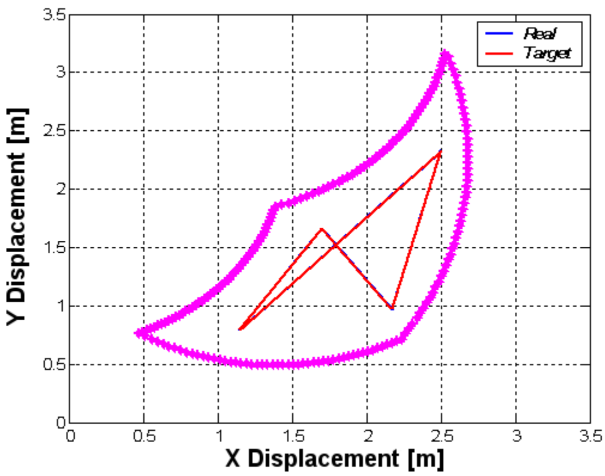

Fig. 25: *Trajectory in the workspace*

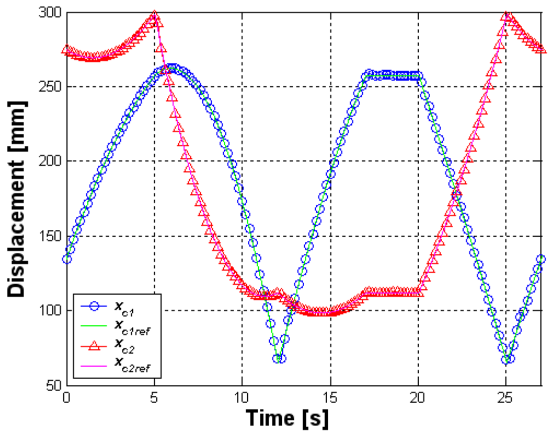

Fig. 26: *Cylinder displacements*

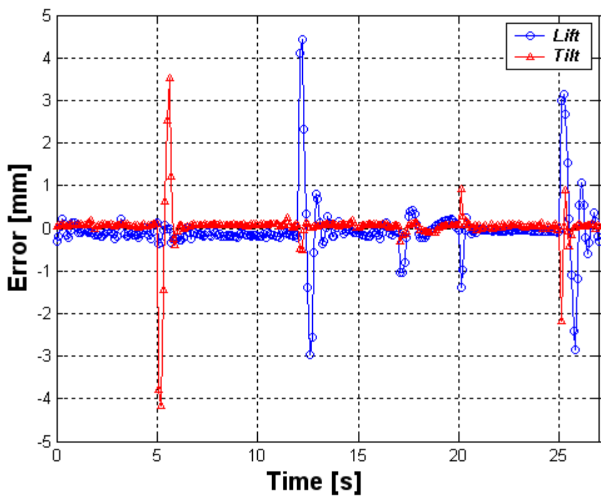

Fig. 27: *Displacement errors*

It must be noticed that the change of direction is sudden, so when the end-effector reaches without deceleration phase the fixed point; it changes direction and velocity with a step. In Fig. 26, the cylinder displacements are shown and in this diagram it is possible to notice this very fast and sudden change of velocity.

These sudden changes of direction influence the value of the errors as shown in Fig. 27.

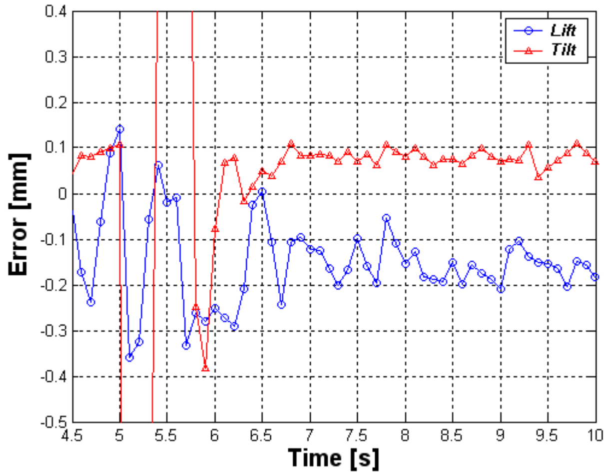

Fig. 28: *Zoom of the displacement errors*

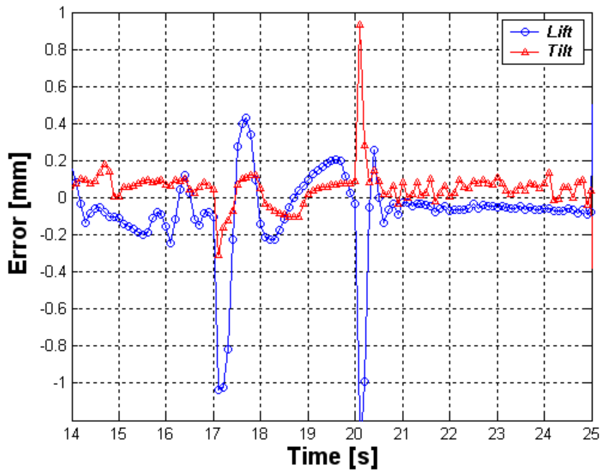

Fig. 29: *Zoom of the displacement errors*

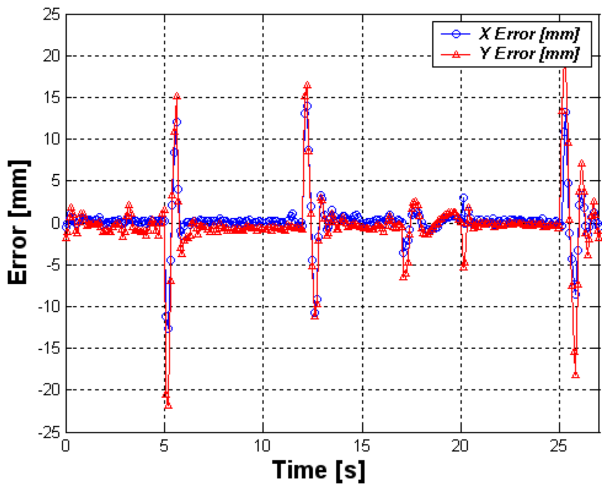

Fig. 30: *Cartesian error*

In fact, when the cylinders must change velocity without any interpolation, the errors due essentially to the inertia of the system, become very high, with a maximum value of about 5 mm for the lift and the tilt too. But these errors have very short time duration, because the controllers can reduce them very quickly and in the most part of the trajectory they are very low as shown in Fig. 28 and 29.

Finally, the Cartesian (positioning errors in the X and Y directions) error are shown in Fig. 30-32.

As noticed for the lift and the tilt displacement errors, the Cartesian errors are normally very low, but they have a sudden increase when the cylinders change direction quickly.

In particular, the Fig. 31 and 32 show a zoom of the Cartesian errors during the sudden change of velocity. Two considerations can be done:

- the errors in most parts of the trajectory is low;
- the effect of the force of gravity causes a Y error greater than the X error.

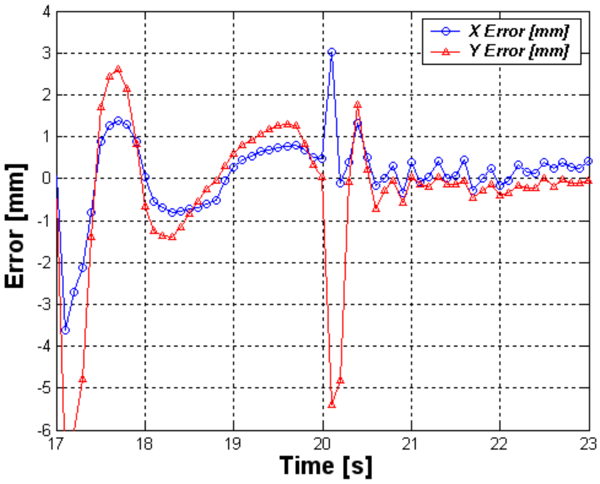

Fig. 31: *Zoom of the displacement errors*

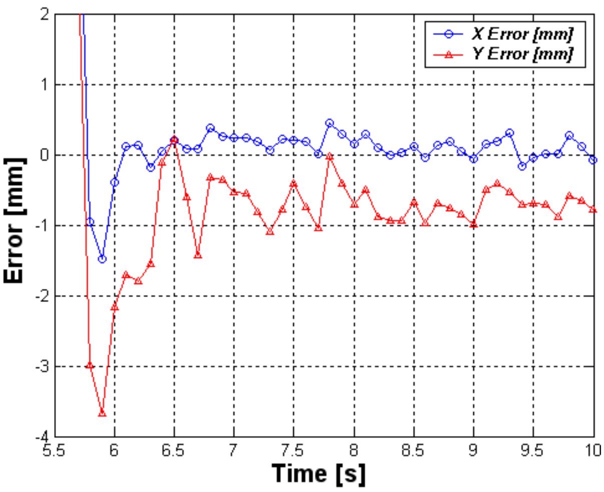

Fig. 32: *Zoom of the displacement errors*

Conclusions

In this work, a commercial crane was equipped by water hydraulic components. A characterisation of the system was realised and the results of the first series of tests were used in order to design an F-PID controller for each actuator.

After the first phase of design and testing of the controllers, a load was put on the crane end-effector and a series of tests have been performed in order to evaluate the performance of the crane in the normal work condition.

Considering the results shown in similar works for oil hydraulic system (Andersen et al., 2005, Bonchis et al., 2001, Saarela et al., 2005) this water hydraulic crane has a very good behaviour and a low level of the positioning error for each actuator. Even by the usage of a very simple technique to control a system geometrically and inertially more complex than a single cylinder, the Water Hydraulic technology allows to obtain a low value of the positioning error moving a considerable mass.

Unfortunately, the low level of the pressure does not allow increasing too much the load on the crane end-effector. For this kind of system the control itself is not the big issue for the future, but the attention must be focused on high pressure components, which can be used in high efficient hydraulic systems and allow to increase the level of the pressure. In this way the Water Hydraulic components will become very competitive and will substitute the Oil Hydraulic components without decreasing of performance.

Nomenclature

A_c	Area piston side	[m^2]
A_v	Opening section of the valve	[m^2]
θ_1	First joint angle	[rad]
θ_2	Second joint angle	[rad]
y_v	Spool position	[m]
x_1	Lift cylinder position	[m]
x_2	Tilt cylinder position	[m]
x_{ref}	Position reference	[m]
v_{ref}	Velocity reference	[m/s]
Q_{ref}	Flow rate reference	[m^3/s]
Δp	Pressure drop	[Pa]
K_P	Proportional gain	[V/m]
K_I	Integrative gain	[V/ms]
K_D	Derivative gain	[Vs/m]
u	Voltage signal	[V]
ρ	Water density	[kg/m^3]

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Simone Bruno
received his Ph.D. in Mechanical Engineering in May 2006. He works in the Department of Mechanical and Industrial Engineering (DIMeG) of the Polytechnic of Bari (Bari, Italy) in the Fluid Power Laboratory with the Prof. A. Lippolis. During his Ph.D. he worked in the Institute of Hydraulic and Automation of the Tampere University of Technology in the research group of the Prof. K.T. Koskinen. His research activity is focused on the study of the dynamic behaviour of hydraulic system.

Harri Sairiala
received his M.Sc. in Mechanical Engineering in 2000. He works as a researcher in the Institute of Hydraulics and Automation (IHA) in Tampere University of Technology (TUT). His research activities are water hydraulic motion control and proportional valves.

Kari T. Koskinen
Professor, Dr.Tech. Kari T. Koskinen is a professor of fluid power in TUT/IHA. He graduated as Dr.Tech. in 1996. Water hydraulics, Virtual technologies in fluid power and aircraft hydraulics belongs to his research interests. He has published over 100 technical and scientific papers. He is leader of the Water Hydraulics Research Group in TUT/IHA.