Non-linear Improvement of an Intelligent Adaptive Controller Designed for Hydraulic Differential Servo Cylinders

József K. Tar*, János F Bitó, Béla Pátkai*** Attila Bencsik+,**

Budapest Polytechnic, *John von Neumann Faculty of Informatics, **Centre of Robotics and Automation, +Donát Bánki Faculty of Mechanical Engineering, H-1081 Budapest, Népszínház utca 8.Hungary
E-mail: {jktar, jbito, bencsik}@zeus.banki.hu
*** Tampere University of TechnologyPOBox 589, Tampere, 33101, Finland
E-mail: bela.patkai@tut.fi

Abstract:
Hydraulic differential, electric servo cylinders are strongly non-linear, coupled multivariable electromechanical tools applicable for driving e.g. manipulators. Besides saturation-type nonlinearities discontinous ones originating from the friction between the cylinder and the piston and the hydrodynamic properties of the flow of the working fluid are present in such systems. Though they have considerable advantages in comparison with electric ones in several applications their traditional PID control have to cope with the problem of instabilities and dificulties in finding the proper setting of their feedback gains. These parameters significantly depend on the actual pose of the robot arm and the temperature of the hydraulic oil varying during operation. To widen their applicability both disturbance rejection based, and partial flatness principle based approaches requiring the accurate model of the system and measurement of the disturbances and their time-derivatives were proposed. For getting rid of needing this accurate set of information an adaptive approach working with a hectic learning phase and an acceptable regime of "tuned" operation was also proposed. In the present paper twoo possible non-linear improvement of this already non-linear control is presented and comaperd to each other. Generally the "learning" phase is shortened and it becomes less hectic, and trajectory tracking becomes more accurate. The capabilities of the improved control are illustrated via simulation.

Keywords: Non-linear Plant; Adaptive Control; Hydraulic Differential Cylinder.

1 Introduction

Hydraulic differential servo cylinders are strongly coupled non-linear electro-mechanical devices of various parameters that are difficult to be kept under perfect control. The viscosity of the oil in the pipe system is very sensitive to the temperature and it normally increases due to the circulation during opration. The oil compressibility depends on the amount of air or other gases solved in it. Solubility of gases in liquids also depends on the pressure and the temperature. Friction like adhesion of the piston at the cylinder introduce roughly non-linear discontinuity into the behavior of the system. Hydrodynamic properties of the flowing oil as well as the not always measurable external disturbance forces can set a quit complex control task.

To control hydraulic differential servo cylinders driven robots a disturbance rejection based approach was proposed in [1]. It required to know an accurate dynamic model of the robot and the real-time measurement of the disturbance forces of external origin and their time-derivative. To reduce the need of such ample amount of accurate information a non-linear adaptive controller was rpoposed in [2]. It was based on a novel concept of soft computing detailed in [3] and realized by the use of the particular algebraic procedure announced in [4]. Like the operation of the cellular neural networks this approaches uses the concept of the *Complete Stability* for the controlled system. The conditions to be met for achieving complete stability were investigated in details in [5] in which it was concluded that in the case of a great variety of non-linear systems complete stability can be achieved. As a particular example a mechanical system driven by hydraulic cylinder was presented in [2]. The aim of this paper is to give further non-linear improvement of this controller in two possible ways. Both approaches use a kind of metrics to measure the significance of the trajectory tracking error. The so measured error infuences the feedback gains of the desired trajectory tracking expressed in purely kinematic terms. The adaptive loops of the controller try to realize this prescribed behavior via machine learning. In the sequel at first the description of the model of the hydraulic differential cylinder and the proposed control is presented, then simulation resulst and conclusions are given.

2 The differential hydraulic cylinder

For modeling the operation of the cylinder the analytic model described by Bröcker in [1] was applied. If x denotes the linear position of the piston in [m] units its acceleration is determined by (1) as

$$\ddot{x} = \frac{1}{m}\left[\left(p_A - \frac{1}{\varphi} p_B\right) A_A - F_f\left(\dot{x}\right) - F_d\right] \tag{1}$$

in which p_A and p_B [*Pa*] denotes the pressures in chamber A and B of the piston, $\varphi=A_A/A_B$ [*non-dimnesional*], that is the ratio of the "active" surfaces of the appropriate sides of the piston, m is the mass of the piston [*kg*], F_f denotes the internal friction between the piston and the cylinder, F_d denotes the external disturbance forces in [*N*] units. The pressure of the oil in the chambers also depends on the piston position as

$$\dot{p}_A = \frac{E_{oil}}{V_A(x)}\left(-A_A\dot{x} + B_v K_v a_1(p_A, U)U\right) \tag{2}$$

$$\dot{p}_B = \frac{E_{oil}}{V_B(x)}\left(\frac{A_A}{\varphi}\dot{x} - B_v K_v a_2(p_B, U)U\right) \tag{3}$$

where B_v denotes the flow resistance, K_v is the valve amplification, U is the *normalized valve voltage*. The oil volume in the pipes and the chambers are expressed as

$$\begin{aligned} V_A(x) &= V_{pipeA} + A_A x, \\ V_B(x) &= V_{pipeB} + A_B(H - x) \end{aligned} \tag{4}$$

(H is the cylinder stroke.) The hydraulic drive has two stabilized pressure values, the *pump pressure* p_0, and the *tank pressure* p_t. Under normal operating conditions (that is when no shock waves travel in the pipeline) these pressures set the upper and the lower bound to p_A and p_B. The functions a_1 and a_2 are defined as given in (5).

$$\begin{aligned} a_1(p_A, U) &= \begin{cases} sign(p_0 - p_A)\sqrt{|p_0 - p_A|} \\ if \quad U \geq 0, \\ sign(p_A - p_t)\sqrt{|p_A - p_t|} \\ if \quad U < 0 \end{cases} \\ a_2(p_B, U) &= \begin{cases} sign(p_B - p_t)\sqrt{|p_B - p_t|} \\ if \quad U \geq 0, \\ sign(p_0 - p_B)\sqrt{|p_0 - p_B|} \\ if \quad U < 0 \end{cases} \end{aligned} \tag{5}$$

If a desired piston acceleration computed on the basis of purely kinematic considerations is needed, on the basis of the available roughly inaccurate and incomplete system model omitting the unknown disturbance force, a *desired value* can be prescribed to (p_A-p_B/φ). Supposing that at least p_A, p_B, x, and dp_A/dt, dp_B/dt, dx/dt are measurable in real-time it is possible to know the *actual value* of this quantity. On this basis a *desired time-derivative* can be prescribed to it. Again, on the basis of an available approximate system model via combination (2) and (3) an appropriate control signal U can be proposed by the controller in order to realize this desired derivative. Here the approach must be self-consistent. First we can suppose that $U>0$. On this basis a_1 and a_2 becomes well defined and U can be calculated from (2) and (3). If the sign of the result accords to the original supposition, this value is used. If not, we use the supposition that $U<0$. (Depending on the modeling errors it may happen that this supposition results in a value for U, which is in contradiction with it. However, because the controller always "has to provide the drives" with some control signal, the consistence of this last supposition is not investigated. Compensation of the consequences of contradiction remains the task of the controller in forthcoming sessions.) Taking into account the inconvenient behavior of the piston's friction, a PI-type controller is applied for (p_A-p_B/φ), and for the desired trajectory tracking, too ("*internal*" and "*external*" adaptive loops).

The adaptive nature of the controller applied consists in the comparison of the *realized* and the *desired* values of the coordinate acceleration and the weighted difference $d(p_A-p_B/\varphi)/dt$, respectively. Via the application of a *dummy parameter* of no direct physical interpretation two two-dimensional vectors are created in which in the 1st row the desired/realized acceleration/time derivative, while in the 2nd row this dummy parameter is located. Following that, exactly as it was proposed in [4], an orthogonal transformation is created which transforms the realized vector into a vector parallel with the desired one while leaving the orthogonal sub-space of these two pairs unchanged. Then proper stretching/shrinking factors are calculated which can make the absolute value of the realized vectors equal to that of the desired ones. On this basis two linear operators are created which apply the appropriate stretches/shrinks in the "*realized*" one-dimensional sub-spaces, rotate them to be parallel to the "*desired*" directions, and leave the orthogonal sub-spaces unchanged. In the next control step the "*desired*" vectors are deformed by these operators before calculating the control signal U. The control has *cumulative nature* in the sense that in the forthcoming steps it is not directly the "*desired*" vector, but the previously already deformed one is further deformed, in both the "external" and the "internal" loops. In the case of convergence in each control step a linear operator is created and the so obtained series has to converge to the identity operator. This procedure means the essence of the adaptive control. (Details of its operation were discussed e.g. in [3] and [5].) Because for the convergence it is needed to work with operators *ab ovo* being in the neighborhood of the identity operator the method is combined with a linear interpolation between the "*desired*" and the "*realized*" vectors to reduce the adaptive controller's burden in the early control steps. In the first steps this reduction is considerable but as the learning process proceeds it becomes negligible.

The above outlined approach worked with fixed PI parameters regarding the prescribed, purely kinematic trajectory tracking. Due to the essential non-linearities of the hydraulic piston/cylinder system too tight desired tracking resulted in undesirable pressure fluctuations in the chambers. In the present approach during the learning phase with drastic errors reduced linear PI gains are applied. As the tracking error is reduced these gains are increased to improve tracking accuracy. The desired racking is described in the “external loop” by

$$\ddot{y}^{Des} = \ddot{y}^{Nom} - K_p\left(y - y^{Nom}\right) - K_i \int \left(y - y^{Nom}\right) dt \tag{6}$$

in which K_p and K_i denotes the proportional and the integral feedback gains. The fixed gains are designed as

$$K_p = \alpha^2,\ K_i = 2\alpha^3 \tag{7}$$

in which α is a typical exponent of the expected error relaxation. Really, the $e{=}exp(-\alpha t)$ error function satisfies the PI-type control

$$\ddot{e} = -\alpha^2 e - 2\alpha^3 \int e dt \tag{8}$$

Due to its noise nature the time-derivative of the error is not applied in the feedback. For varying α also (7) was used, with the law of variation

$$\alpha = \alpha_0 \left(2.2 - \frac{60\xi}{1 + 60\xi} \right) \tag{9}$$

in which parameter ξ measures th significance of the error, and $0<\alpha_0$ is constant. Near zero error results in strong feedback gains, large error results in small one. For ξ different measures can be applied. The simplest one can be

$$\xi := \left| y^{Nom} - y \right| \tag{10}$$

just taking into account the actual trajectory reproduction error. A more sophisticated choice can measure the absolute value of the *error,* the *speed of error,* and the *integrated error* as

$$\xi = \left(c_0 \left| \dot{y}^{Nom} - \dot{y} \right| + \alpha_0^2 \left| y^{Nom} - y \right| + 2\alpha_0^3 \left| \int \left(y^{Nom} - y \right) dt \right| \right) / 1000. \tag{11}$$

The positive coefficients (c_0>0, too) in (11) represent some metric due to the absolute values to which they are related which seems to be appropriate to describe the tracking error of a non-linear plant. Indeed, it is relevant to suppose in the case only of a strictly linear plant that a *large positive actual error* to some extent is compensated by a *large negative change in the error*, etc.. Such typical suppositions often used in constructing *fuzzy rules* hide the latent supposition that the plant behaves in a more or less *linear way*, and the task of a fuzzy controller is to “slightly” adjust the feedback gains. However, in the case of a non-linear plant the combination of the trems with their signs as given in (8) not necessarily conveys satisfactory information on the measure of the error because its different constituents as they are given in (11) may have quite specific action. I the sequel a typical non-linear plant, the 1 DOF robot arm driven by a hydraulic differential cylinder is investigated.

2 Simulation results

Unfortunately no satisfactory room we have to give all the detailed parameters of the system. The most important is to note that we supposed that the compressibility of the hydraulic oil used is considerably reduced by the application of appropriate elastic elements built in the system. Too small compressibility could mak the system too stiff to control. In the simulations 20% error is applied in modeling this reduced oil compressibility. The piston's friction is taken into account in simulating the real system by the Stribeck model, but it is unknown for the controller. A constant external disturbance force (unknown by the controller) of 500 *N* also was applied in the simulations.

In order to give some idea about the significance of model uncertainties and unknown external dynamic forces in Fig. 1 some characteristics of the non-adaptive, the internally adaptive only, and the externally adaptive motion are presented. It can well be seen that the internal adaptive loop decreases the pressure fluctuation in the chamber and results in a smooth motion with considerable tracking error. Application only the external adaptive loop cannot eliminate the pressure fluctuations.

In Fig. 2 the phase space and trajectory tracking of the fully adaptive motion is given according to Eq. (10), Eq. (11) with c_0=0, and Eq. (11) with $c_0=0.1\times\alpha_0$, respectively. It is clear that the latter one gives the best result though the difference between the last two solutions is quite small.

To reveal internal details of the control the change of the pressure in the chambers 'A' and 'B' of the differential cylinder are described in Fig. 3. The fluctutaion of the chamber pressures is kept at bay due to the adaptivity.

In the simulations the stretch/shrink operations in El Hini's transformations [4] remained practically equal to 1. The angle of abstract rotation in [*rad*] units is the parameter really characteristic to the necessary transformations. According to Fig. 4 in each case we have very small cumulative (or with other words consecutive) rotations. Due to the discontinuous friction in the cylinder near the settling point of zero velocity considerable fluctuation of the friction forces can be observed in Fig. 4, too. From this point of view the definitely non-zero velocity parts of the nominal trajectory mean far less challenge for the controller than parts at which near zero elocity is needed and a slight fluctuation of the sign of the velocity causes drastic change in the perturbation force due to the non-linear coupling caused by the friction.

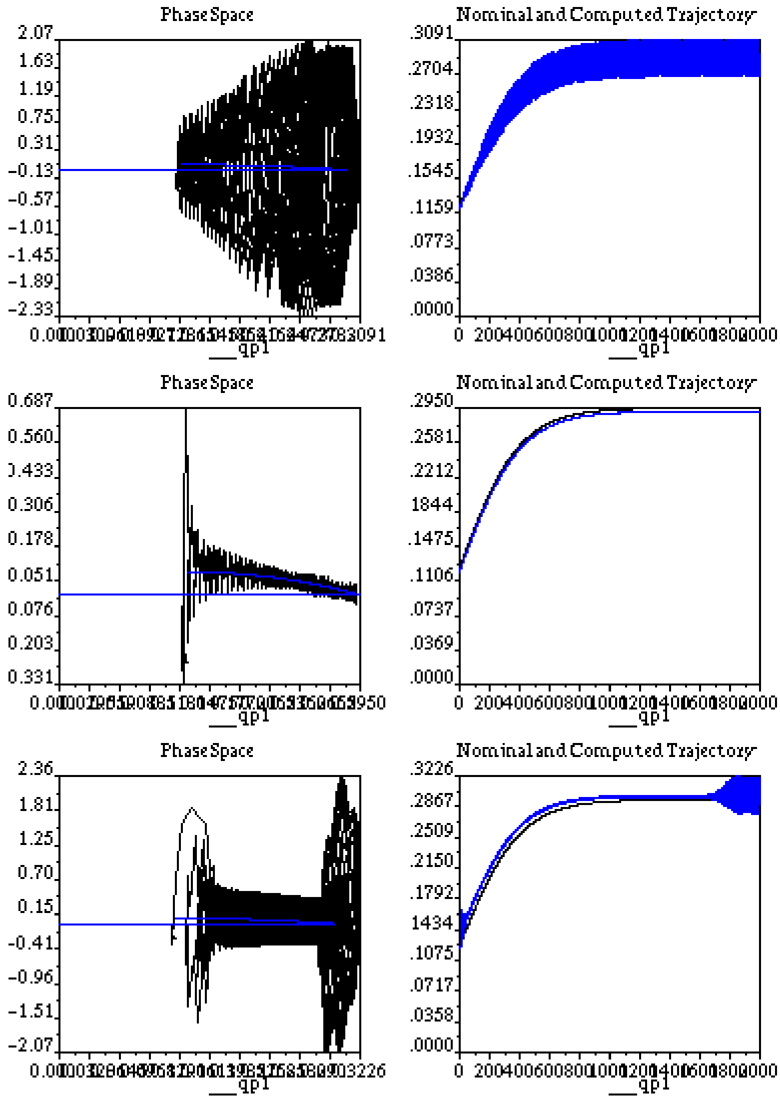

Fig. 1. Phase space ([*m*], [*m/s*]) and trajectory tracking [*m*] for the non-adaptive, internally adaptive only, and externally adaptive only control, time in [*ms*] units.

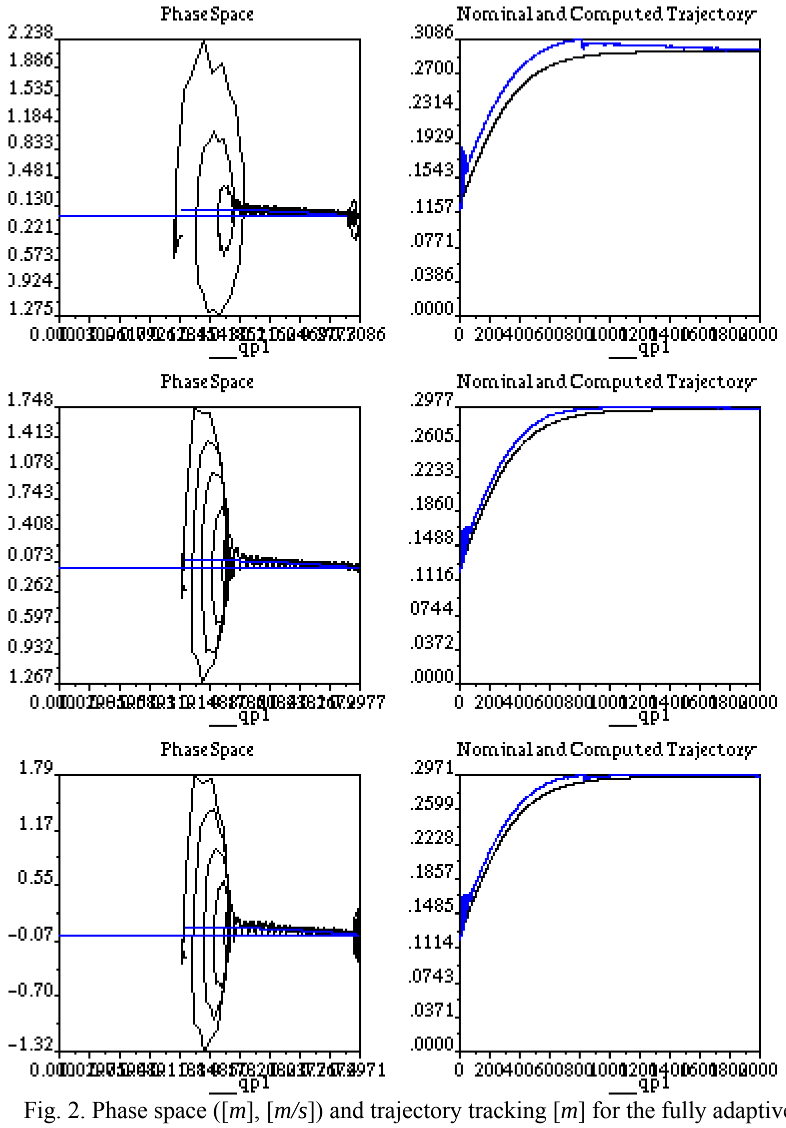

Fig. 2. Phase space ([*m*], [*m/s*]) and trajectory tracking [*m*] for the fully adaptive cases according to Eq. (10), Eq. (11) with c c_0=0, and Eq. (11) with c_0=0.1×α_0, respectively

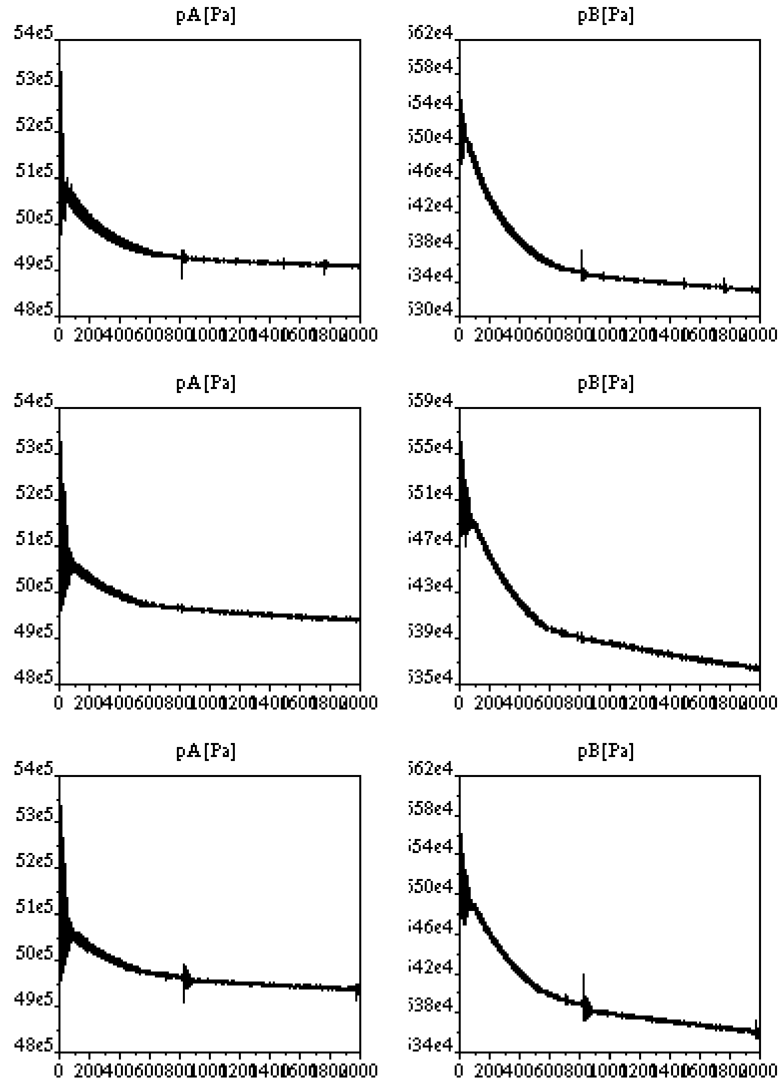

Fig. 3. Chamber pressures [Pa] for the fully adaptive cases according to Eq. (10), Eq. (11) with c c_0=0, and Eq. (11) with c_0=0.1×α_0, respectively, time in [ms] units.

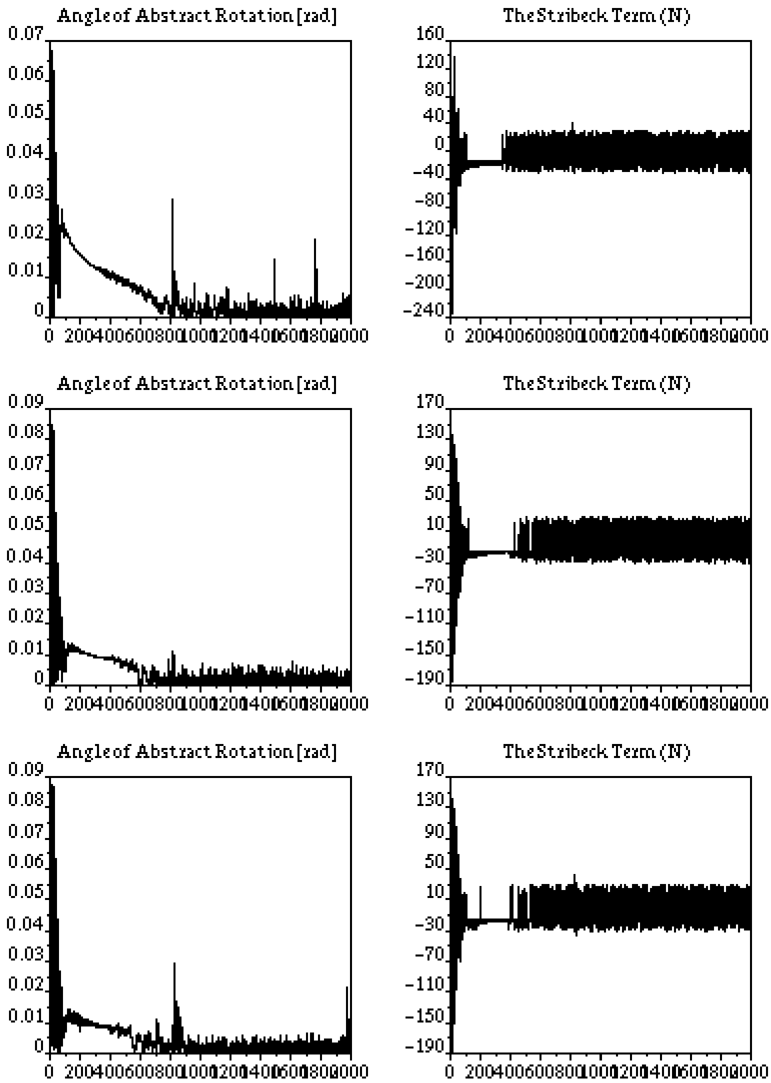

Fig. 4: El Hin's abstract rotations [*rad*] and the friction forces in the piston [*N*] for the fully adaptive cases according to Eq. (10), Eq. (11) with c c_0=0, and Eq. (11) with c_0=0.1×α_0, respectively, time in [*ms*] units.

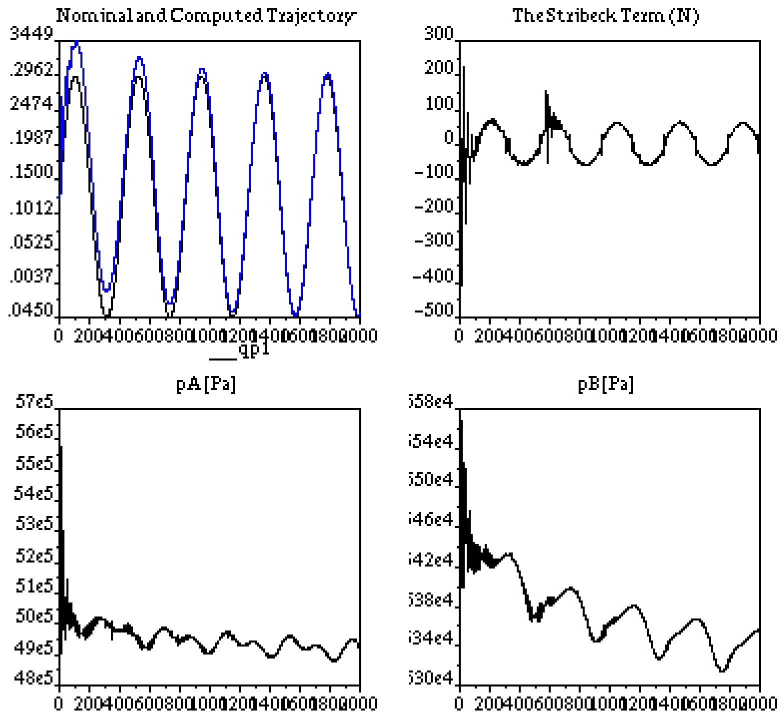

Fig. 5. Trajectory tracking [*m*], the friction forces in the piston [*N*], and the variation of the chamber pressures [*Pa*] for the fully adaptive case according to Eq. (11) with $c_0=0.1\times\alpha_0$, for periodic nominal motion, time in [*ms*] units.

To illustrate that in Fig. 5 the trajectory tracking, the friction forces, and the chamber pressures are described for periodic nominal trajectory. In this case the adhesive friction causes only one jump in the friction force when the velocity of the piston changes sign.

4 Conclusions

In this paper two kinds of improved non-linear version of a recently developed adaptive controller was studied in the control of a differential hydraulic cylinder under external disturbances not modeled by the controller. The simulation results illustrate well that such an adaptive technique can improve the controller's operation. The information need of the adaptive controller is far much less than that of the traditional disturbance rejection controller needing the measurement of the disturbance force and its time-derivative and the accurate model of the drive. In spite of the quite drastic and even discontinuous nonlinearities of the plant to be controlled the method promises good operation and its further development and investigation seems to be reasonable.

5 Acknowledgment

The authors gratefully acknowledge the support by the Hungarian-Polish S&T PL-2/01 project for 2002-2003, and that of the Hungarian National Research Fund (OTKA T034651, T034212 projects).

6 References

[1] M. Bröcker, M. Lemmen: "Nonlinear Control Methods for Disturbance Rejection on a Hydraulically Driven Flexible Robot", in the Proc. of the Second International Workshop On Robot Motion And Control, RoMoCo'01, October 18-20, 2001, Bukowy Dworek, Poland, pp. 213-217, ISBN: 83-7143-515-0, IEEE catalog Number: 01EX535.

[2] József K. Tar, Marcus Bröcker, Krzysztof Kozlowski: “A Novel Adaptive Control for Hydraulic Differential Cylinders”, in the Proc. of the 11th Workshop on Robotcs in Alpe-Adria-Danube Region, June 30 - July 2, 2002, Balatonfüred, Hungary, ISBN: 963 7154 10 8 (for the issue on CD), pp. 7-12.

[3] J.K. Tar, M. Rontó: "Adaptive Control Based on the Application of Simplified Uniform Structures and Learning Procedures", Zbornik Radova, Vol. 24 No. 2, 2000, pp. 174-194. (ISSN: 0351-1804.

[4] Yahya El Hini: "Comparison of the Application of the Symplectic and the Partially Stretched Orthogonal Transformations in a New Branch of Adaptive Control for Mechanical Devices", Proc. of the 10th International Conference on Advanced Robotics", August 22-25, Budapest, Hungary, pp. 701-706, ISBN 963 7154 05 1.

[5] József K. Tar, János F. Bitó, Krzysztof Kozłowski, Béla Pátkai, D. Tikk: "Convergence Properties of the Modified Renormalization Algorithm Based Adaptive Control Supported by Ancillary Methods", in the Procthe 3rd International Workshop on Robot Motion and Control (ROMOCO ’02), Bukowy Dworek, Poland, 9-11 November, 2002, pp. 51-56, ISBN 83-7143-429-4, IEEE Catalog Number: 02EX616.