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INTEGRATION OF SOFT COMPUTING AND FRACTIONAL DERIVATIVES IN ADAPTIVE CONTROL

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Abstract. Realizing that generality and uniformity of the usual Soft Computing (SC) structures exclude the application of plausible simplifications relevant in the case of whole problem classes resulted in the idea that a novel branch of soft computing could be developed by the use of which far simpler and more lucid uniform structures and procedures could be applied than in the traditional ones. Such a novel approach to computational cybernetics akin to SC was developed at Budapest Tech to control inaccurately and incompletely modeled dynamic systems under external disturbances. Hydraulic servo valve controlled differential cylinders as non-linear, strongly coupled multivariable electromechanical tools serve as excellent paradigms of such difficulties. Their control has to cope with the problem of instabilities due to the friction forces between the piston and the cylinder, as well as with uncertainties and variation of the hydrodynamic parameters that makes it unrealistic to develop an accurate static model for them. In this paper a combination of this novel method with the use of fractional derivatives is applied for the control of a hydraulic differential cylinder. Simulation results well exemplifying the conclusions are also presented.

Keywords: Soft computing, uniform structures and procedures, adaptive control, hydraulic differential cylinders

1 INTRODUCTION

The main advantage in using SC is the evasion of the development of intricate analytical, static models of the systems to be controlled. Its fundamental components were almost completely developed by the sixties of the past century. In our days SC means either separate or integrated application of Neural Networks (NN) and Fuzzy Systems (FS) enhanced with high parallelism of operation and supported by several deterministic, stochastic or combined parameter-tuning methods that are frequently referred to as "learning".

Regarding the use of NNs, typical problem classes have been identified for the solution of which typical uniform architectures (e.g. multilayer perceptron, Kohonen-network, Hopfield-network, Cellular Neural Networks, CNN Universal Machine, etc.) have been elaborated. For instance, a typical application of NNs is the linearization of nonlinear sensor signals [1].

The great advantage in the use of fuzzy systems is that they provide mathematically rigorous representation of vague or imprecise information in a form similar to human languages [2]. They use membership functions of typical (e.g. trapezoidal, triangular or step-like, etc.) shapes, and the fuzzy relations can also be utilized in a standardized way by using different classes of fuzzy operators.

The first phase of applying traditional SC, that is the identification of the problem class and finding the appropriate structure for dealing with it, is easy normally. The next phase, i.e. determining the necessary size of the structure and fitting its parameters via machine learning, is far less easy. In general, in the case of strongly coupled non-linear Multiple Input – Multiple Output (MIMO) systems traditional SC suffers from the disadvantage of wrong "scalability" or "curse of dimensionality" that mean that the number of the necessary neurons/fuzzy rules strongly increases with the degree of freedom and the intricacy of the problem.

To reduce modeling complexity fuzzy interpolation methods were developed and checked. For instance, similarity relations can be utilized in the design of fuzzy diagnostic systems [3]. Various techniques were elaborated as remedies for such problems as the application of rule interpolation [4] or improving Sugeno and Yasukawa's Qualitative Modeling [5], or the application of hierarchical rules [6], etc. In spite of the very important developments the situation in such approaches normally is not lucid. Similar problems arise regarding the necessary number of neurons in a neural network approach. External dynamic interactions on which normally no satisfactory information is available influence the system's behavior in dynamic manner. Both the big size of the necessary structures, the huge number of parameters to be tuned, as well as the goal varying in time still are serious problems.

In order to get rid of the scalability problems of the classical Soft Computing a novel approach was initiated on the basis of a compromise between the need of generality and scalability in [7]. It was shown by the use of perturbation calculus that this method can be applied for a quite wide class of physical systems, e.g. in the case of classical mechanical systems, too [8]. This approach uses far simpler and far more lucid uniform structures and procedures than the classical ones: various algebraic blocks originating from different Lie groups can be incorporated into the "model", e.g. a new family of symplectic transformations [9]. In the present paper this method is applied in adaptive control of an electromagnetic servo valve controlled differential hydraulic cylinder first investigated by Bröcker and Lemmen [10].

Their first approach was based on the "disturbance rejection principle", the other one on the "partial flatness principle". In each case *it was necessary to measure the disturbance force and its time-derivative as well as to know the exact model of the hydraulic cylinder they developed in details and identified for a particular robot arm-drive system.* However, the identification of such a system needs a lot of laboratory work the result of which may also be temporal. A serious problem is the need for measuring the external disturbance forces. In general it seems to be expedient to apply adaptive control instead of trying to measure the ample set of unknown and time-varying parameters. However, this adaptive control need not be too intricate, actually should not be much more complicated than an industrial PID controller. For this purpose Soft Computing based approaches would be more attracting than detailed analytical modeling.

The above approach was applied in adaptive control of servo valve controlled differential hydraulic cylinder in [11]. This approach used the phenomenology of the hydraulic cylinder in a very cautious manner that avoided prescribing a PID control due to the general angst regarding the non-linearities generated by friction. A PI control was applied for the piston's trajectory only. The approach described in this paper transcends this previous one by allowing a PID^{var} control for the piston's trajectory, in which the order of derivation depends on the past fluctuation of the piston's velocity that generates harsh modification in the friction forces, especially in the vicinity of the zero-transitions of the velocity.

In the sequel the main point of the scalable soft computing is very briefly outlined. Following that the analytical model of the differential hydraulic servo cylinder is presented together with the new control approach applied. The paper is closed by the simulation results and the conclusions.

2 FORMULATION OF THE CONTROL TASK

From purely mathematical point of view the control task can be formulated as follows. Some imperfect model of the system is given, on the basis of which some excitation is calculated to obtain a desired system response $\mathbf{i}^d$ as $\mathbf{e} = \varphi(\mathbf{i}^d)$. The system has its inverse dynamics described by the unknown function $\mathbf{i}^r = \psi(\varphi(\mathbf{i}^d) = \mathbf{f}(\mathbf{i}^d)$ and resulting in a realized response $\mathbf{i}^r$ instead of the desired one, $\mathbf{i}^d$. Normally one can obtain information via observation only on the function $\mathbf{f}()$ considerably varying in time, and no possibility exists to directly "manipulate" the nature of this function: only $\mathbf{i}^d$ as the input of $\mathbf{f}()$ can be "deformed" to $\mathbf{i}^{d^*}$ to achieve and maintain the $\mathbf{i}^d = \mathbf{f}(\mathbf{i}^{d^*})$ state. On the basis of the modification of the method of renormalization widely applied in physics the following "iteration" was suggested for finding the proper deformation:

$$\begin{gathered}\mathbf{i_0};\ \mathbf{S}_1\mathbf{f}\left(\mathbf{i_0}\right) = \mathbf{i_0};\ \mathbf{i}_1 = \mathbf{S}_1\mathbf{i_0};\ \dots;\ \mathbf{S}_\mathrm{n}\mathbf{f}\left(\mathbf{i}_{n-1}\right) = \mathbf{i_0};\\ \mathbf{i}_{n+1} = \mathbf{S}_{n+1}\mathbf{i}_n;\ \mathbf{S}_\mathrm{n} \xrightarrow[n\to\infty]{} \mathbf{I}\end{gathered} \tag{1}$$

in which the $\mathbf{S}_n$ matrices denote some linear transformations. These matrices map the observed response to the desired one, and the construction of each matrix corresponds to a step in the adaptive control. It is evident that if this series converges to the identity operator just the proper deformation is approached, therefore the controller "learns" the behavior of the observed system by step-by-step amendment and maintenance of the initial model.

Regarding the resolution of the ambiguity of the matrices mathematically not fully defined by (1), evasion of the dubious $0 \to 0$, $0 \to$finite, finite$\to 0$ transformations, and the convergence of the method, we refer to [7] and [9].

3 DESCRIPTION OF THE SYSTEM TO BE CONTROLLED

The operation of the differential hydraulic cylinder was described in details e.g. in [10]. Let x denote the linear position of the piston in m units. The acceleration of the piston is described by [10] as

$$\ddot{x} = \frac{1}{m}\left[\left(p_A - \frac{1}{\varphi}p_B\right)A_A - F_f\left(\dot{x}\right) - F_d\right] \tag{2}$$

in which p_A and p_B denote the pressures in chambers A and B of the piston in bar units. $\varphi = A_A/A_B$ denotes the ratio of the "active" surface areas of the appropriate sides of the piston, m is the mass of the piston in kg, F_f stands for the internal friction between the piston and the cylinder, and F_d denotes the external disturbance force. The pressure of the oil in the chambers also depends on the piston position and velocity as

$$\dot{p}_A = \frac{E_{oil}}{V_A\left(x\right)}\left(-A_A\dot{x} + B_vK_va_1\left(p_A, sign\left(U\right)\right)U\right) \tag{3}$$

$$\dot{p}_B = \frac{E_{oil}}{V_B\left(x\right)}\left(\frac{A_A}{\varphi}\dot{x} - B_vK_va_2\left(p_B, sign\left(U\right)\right)U\right) \tag{4}$$

where B_v denotes the flow resistance, K_v is the valve amplification, U is the *normalized valve voltage*.

The oil volumes in the pipes and in the chambers are expressed as

$$\begin{aligned} V_A(x) &= V_{pipeA} + A_A x, \\ V_B(x) &= V_{pipeB} + A_B(H - x) \end{aligned} \tag{5}$$

(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 (6). Under "normal conditions" $sign(a_1) \geq 0$, and $sign(a_2) \geq 0$, too, according to the limiting role of the pump and tank pressures.

$$\begin{aligned} a_1(p_A, sign(U)) &= \begin{cases} sign(p_0 - p_A)\sqrt{|p_0 - p_A|} \\ if \quad U \geqslant 0, \\ sign(p_A - p_t)\sqrt{|p_A - p_t|} \\ if \quad U < 0 \end{cases} \\ a_2(p_B, sign(U)) &= \begin{cases} sign(p_B - p_t)\sqrt{|p_B - p_t|} \\ if \quad U \geqslant 0, \\ sign(p_0 - p_B)\sqrt{|p_0 - p_B|} \\ if \quad U < 0 \end{cases} \end{aligned} \tag{6}$$

The phenomenology on the basis of which the control in [11] was formed sounds as follows. Supposing the need for a desired piston acceleration computed on the basis of purely kinematic considerations, on the basis of the available system model (either omitting or involving terms regarding the piston friction), and omitting the unknown disturbance force, a *desired value* can be prescribed to $(p_A - p_B/\varphi)$. Supposing that at least p_A, p_B, x, and $\mathrm{d}p_A/\mathrm{d}t$, $\mathrm{d}p_B/\mathrm{d}t$, $\mathrm{d}x/\mathrm{d}t$ 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 the linear combination of equations $[(3)\text{-}(4)/\varphi]$ an appropriate control signal U can be proposed by the controller in order to realize this desired derivative. Due to the inconvenient behavior of the piston's friction, a PI-type controller was proposed for $(p_A - p_B/\varphi)$, and for the desired trajectory tracking, too. Via comparing the desired and the observed values the adaptive control briefly outlined above was applicable for this problem. In this manner the derivation of the disturbance force was evaded in contrast to Bröcker's original approach in [11] that started with the derivation of (2) according to the time.

In the present paper an alternative approach was chosen. For the desired relaxation of the trajectory tracking error $e := (x^R - x^{Nom})$ a simple kinematic formula was prescribed as

$$\ddot{e} = -Pe - D\dot{e} - I\int_0 e\,\mathrm{d}t. \tag{7}$$

With properly chosen P, I, and D coefficients (7) evidently corresponds to the mixture of error components of exponential damping. The appropriate coefficients were determined simply by substituting an expected $e = \exp(\alpha t)$ type relaxation into the time-derivative of (7) that resulted in a third order polynomial for α. For this polynomial three different negative real roots were prescribed in the form of $(\alpha - \alpha_1)(\alpha - \alpha_2)(\alpha - \alpha_3)$. Substituting this into (7) the proper P, D, and I parameters were conveniently determined. The time-derivative of (7) therefore lead to the *desired third time-derivative of the piston's trajectory* as

$$\dddot{x}^d = \dddot{x}^{Nom} - P\dot{e} - D\ddot{e} - Ie. \tag{8}$$

The very rough approximate model of the cylinder was obtained by omitting the friction forces and the external disturbance forces in (2) as

$$\dddot{x}^d = \frac{A_A}{m}\left(\dot{p}_A - \frac{1}{\varphi}\dot{p}_B\right) \tag{9}$$

into which the desired time-derivative of the piston's acceleration was substituted. Equation (9) thus immediately yields an expected value for $\mathrm{d}(p_A - p_B/\varphi)/\mathrm{d}t$. Via computing $[(3)\text{-}(4)/\varphi]$ the proposed control signal U can be determined, and from the known current state of the system from (3), and (4), $\mathrm{d}p_A/\mathrm{d}t$, and $\mathrm{d}p_B/\mathrm{d}t$ can be determined. Consider now the Euler-Lagrange equation of motion of the classical mechanical systems and its time-derivative in a wider context!

$$\mathbf{M}(\mathbf{q})\ddot{\mathbf{q}} + \mathbf{C}(\mathbf{q}, \dot{\mathbf{q}}) = \mathbf{Q}, \tag{10}$$

$$\mathbf{M}(\mathbf{q})\dddot{\mathbf{q}} + \dot{\mathbf{M}}(\mathbf{q}, \dot{\mathbf{q}})\ddot{\mathbf{q}} + \dot{\mathbf{C}}(\mathbf{q}, \dot{\mathbf{q}}) = \dot{\mathbf{Q}} \tag{11}$$

It is evident from (10) that if $\mathrm{d}^2\mathbf{q}/\mathrm{d}t^2$ does not suffer abrupt variation, for abrupt variation of $\mathrm{d}\mathbf{Q}/\mathrm{d}t$ abrupt variation of $\mathrm{d}^3\mathbf{q}/\mathrm{d}t^3$ can be expected, that is the strictly positive definite inertia matrix of the mechanical system

$$\frac{\partial\dot{\mathbf{Q}}}{\partial\dddot{\mathbf{q}}} = \frac{\partial\mathbf{Q}}{\partial\ddot{\mathbf{q}}} = \mathbf{M} \tag{12}$$

sets similar relationship between $\mathrm{d}\mathbf{Q}/\mathrm{d}t$ and $\mathrm{d}^3\mathbf{q}/\mathrm{d}t^3$ and between $\mathbf{Q}$ and $\mathrm{d}^2\mathbf{q}/\mathrm{d}t^2$. In the proof of the possibility of convergence of (1) for mechanical systems this property of $\mathbf{M}$ was utilized when the desired/realized values were prescribed/observed for $\mathrm{d}^2\mathbf{q}/\mathrm{d}t^2$. Therefore the possibility for convergence still exists if, on the basis of (11), the desired/realized values are prescribed/observed for $\mathrm{d}^3\mathbf{q}/\mathrm{d}t^3$, when in the role of the physical agent controlling the motion $\mathrm{d}\mathbf{Q}/\mathrm{d}t$ stands instead of $\mathbf{Q}$. With appropriate initial conditions, in the field of classical mechanics in general (11), and in the particular case of the piston now considered, (9) just corresponds to this situation. In the present paper this observation is utilized. In connection with its applicability some attention has to be paid to the problem of observing $\mathrm{d}^3x/\mathrm{d}t^3$, which, in the case of the presence of friction forces, may be critical. For filtering the noisy part of this signal, in accordance with the concept of fractional order derivatives Caputo's definition can be applied, that *re-integrates* the integer order derivative with a kernel function of long tail acting as a frequency filter. The method was found to be promising for damping the forced oscillation of a car chassis while passing hills/valleys along a bumpy road [12]. According to that (8) can be modified as

$$x^{(2+\beta)d} = \int\limits_0^t d\tau \left[\dddot{x}^{Nom}(\tau) - P\dot{e}(\tau) - D\ddot{e}(\tau) - Ie(\tau)\right] \times \frac{(t-\tau)^{-\beta}}{\Gamma(1-\beta)},\ \beta \in (0,1)\,. \tag{13}$$

In the practical realization of (13) the lower limit of the integration is replaced – $t-T$ instead of 0 – that corresponds to a finite "memory of length" T. Furthermore, for the numerical approximation of the integral with singular integrand the following formula can be used: the full interval of the integration of length T is divided into small sub-intervals of length δ, during which the reintegrated derivative is supposed to be approximately constant:

$$\frac{\mathrm{d}^\beta}{\mathrm{d}t^\beta}u(t) \cong \frac{u'(t)\,\delta^{-\beta+1}}{\Gamma(2-\beta)} + \sum_{0<s<T/\delta} \frac{\delta^{-\beta+1}\left[(s+1)^{-\beta+1} - s^{-\beta+1}\right]}{\Gamma(2-\beta)} u'(t-s\delta). \tag{14}$$

The next essential point is setting the order of derivation depending on the behavior of the velocity of the piston. As is well known changing sign of the velocity generates drastic changes in the friction forces. In the Stribeck model of friction the viscous friction forces and adhesion are combined (details are given e.g. in [10]). Due to the controller's feedback this friction force can oscillate whenever zero-transmission happens in the velocity. That is, $\beta \approxeq 1$ is needed for non-zero velocities, and $\beta < 1$ whenever the velocity is in the vicinity of zero. In the present paper the following adaptive formula was applied

$$0 < \beta = \frac{A + \left|\sum\limits_{s=1}^{T/\delta} sign\left(\dddot{x}(t-s\delta)\right)\right|^\gamma}{A + (T/\delta)^\gamma} \leqslant 1 \tag{15}$$

in which instead of the velocity the observed 3^{rd} time-derivatives are used, because this signal is directly related to the controller's feedback. In the forthcoming simulations the numerical data of the system measured and published by Bröcker and Lemmen in [10] were used, with the exception of the oil elasticity E_{oil} for which Bröcker measured $1\,800 \times 10^6$ Pa, which is a huge value representing the approximate incompressibility of the liquid. However, in a pipe system, due to elasticity of the pipe walls, or due to complementary components intentionally built into the system to reduce this huge stiffness (e.g. via using hydraulic accumulators, flexible hoses [13]) this value can be considerably smaller. In the sequel simulation results are presented.

4 SIMULATION RESULTS

In the initial simulations $E_{oil} = 18 \times 10^6\ Pa$ and $f_{\nu i} = 175\,\mathrm{Nsm}^{-1}$. The other significant constants in (14) were determined via running various simulations, and had the values as follows: $\delta = 1\,\mathrm{ms}$, $T = 20\,\mathrm{ms}$ (also corresponds to the cycle time of the external adaptive loop using the symplectic transformations), $A = 1$, $\gamma = 10^{-4}$ *nondimensional*. As a disturbance force a constant 500 N and a sinusoidal force of amplitude 200 N and circular frequency $12\,\mathrm{s}^{-1}$ was applied.

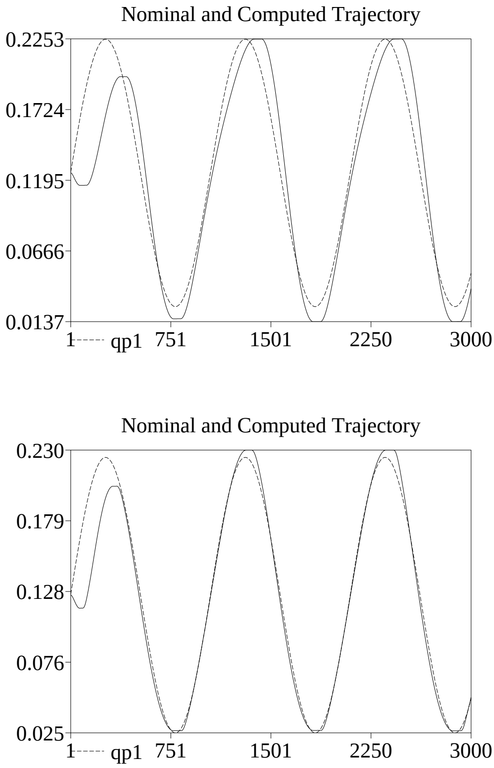


Fig. 1. Nominal and simulated piston trajectory [m] vs. time [ms] for the PID^{var} (upper), and the combined (PIDvar and symplectic transformations) (lower) controls

In Figure 1 the trajectory tracking properties of the controller without and with the application of the external adaptive control loop can be seen. The improvement is evident. Figure 2 reveals that the variation of the pressure in the chambers of the cylinder is not "hectic", in spite of the hectic behavior of the friction forces in the vicinity of the zero transition of the piston's velocity. It is interesting to see what happens whenever the nominal trajectory asymptotically is "constant", i.e. when the region of the friction force's instability lasts in time. Figure 3 displays typical results for this case in which the information necessary for the external adaptive loop becomes dubious for a long time, and the simple PID^{var} control seems to be even a little bit better than the full one.

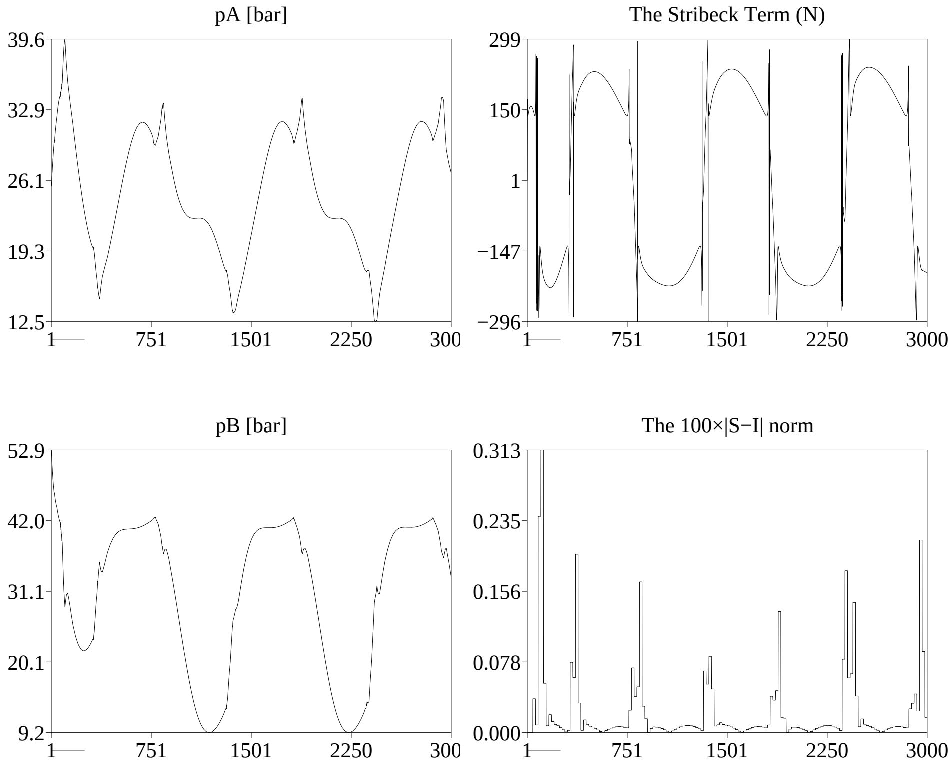


Fig. 2. The variation of the pressure in the cylinder chambers, the friction forces, and the "extent" of the external adaptive transformation vs. time [ms] for the fully adaptive control

In order to combine the sinusoidal and the asymptotically constant desired trajectories, in Figures 4 and 5 simulation results can be seen for the original viscosity $f_{\nu i} = 175\,\mathrm{Nsm}^{-1}$ and its increased value $f_{\nu i} = 400\,\mathrm{Nsm}^{-1}$. (The other parameters of the simulation were not varied.) It is evident that in the high velocity part the tracking accuracy decreased to some extent, while in the low velocity parts the effect of the viscosity is not significant in comparison with that of the adhesion of the piston. The phase trajectories reveal that no extreme acceleration happens. As it was expected in the less critical parts of the motion the order of derivation is integer (≈ 3), and it sharply decreases in the close vicinity of the critical points (Figure 5).

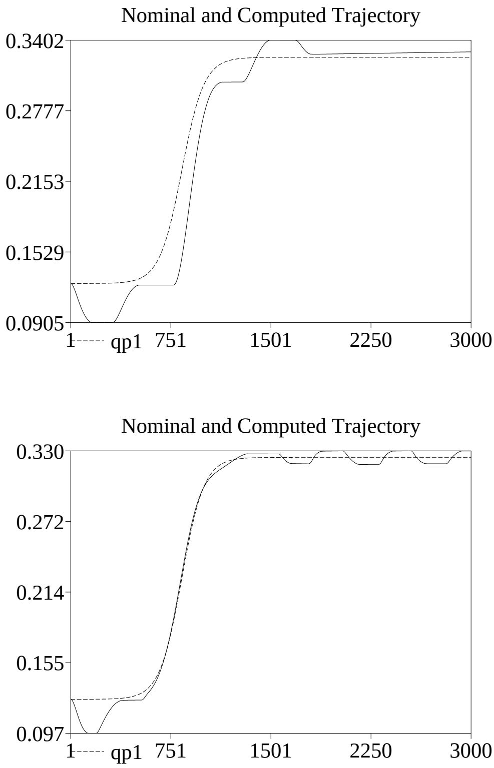


Fig. 3. Tracking properties of the the simple PID^{v}ar and the fully adaptive control (piston friction included): piston position [m] vs. time [ms]

5 CONCLUSIONS

In this paper a possible improvement of an adaptive control developed for magnetic servo valve operated differential hydraulic cylinders was considered. It was found that whenever the piston is in a "cruising phase" of nonzero velocity, simultaneous application of tuning the order of differentiation and that of a special external adaptive loop results in considerable achievement. For this success it is necessary to increase the net compressibility of the hydraulic working liquid by the application of accumulators. In the case of trajectories having asymptotically zero velocity the piston's friction still remains a considerable disturbing factor. In the asymptotical part of these trajectories the simple PID^{var} control seems to be more advantageous.

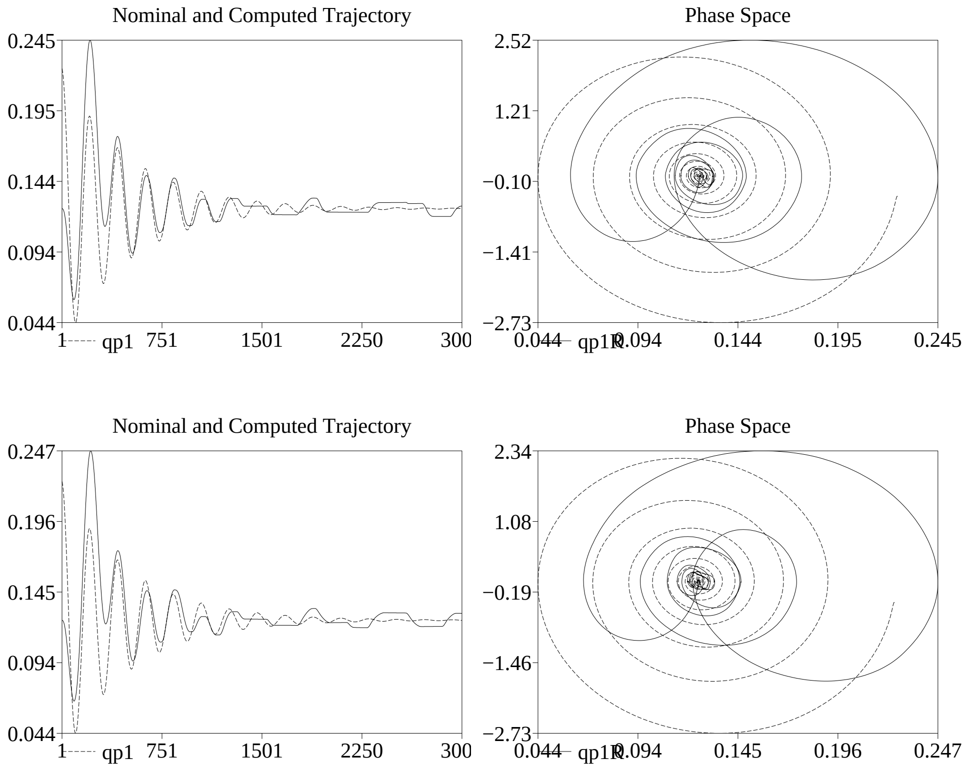


Fig. 4. Trajectory (x in m vs. time in ms) and phase trajectory ($\mathrm{d}x/\mathrm{d}t$ in m/s vs. x in m) tracking for the fully adaptive controller for the original (upper pair) and the increased (lower pair) viscosity

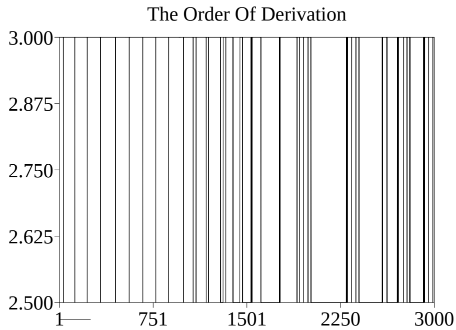
REFERENCES

[1] Kováčová, I.—Madarász, L.—Kováč, D.—Vojtko, J.: Proc. of the 8^{th} IEEE International Conference on Intelligent Engineering Systems. ISBN 973-662-120-0, pp. 79–82.

[2] Vaščák, J.—Madarász, L.: Bulletin for Applied Mathematics. Technical University of Budapest and Technical University of Košice, October 1995, pp. 247–250.

[3] Vaščák, J.—Madarász, L.: Journal of Advanced Computational Intelligence. Vol. 4, 2001, No. 4, pp. 246–250.

[4] Tikk, D.—Baranyi, P.—Gedeon, T. D.—Muresan, L.: Tatra Mountains Math. Publ. 2001, Vol. 21, pp. 73–91.



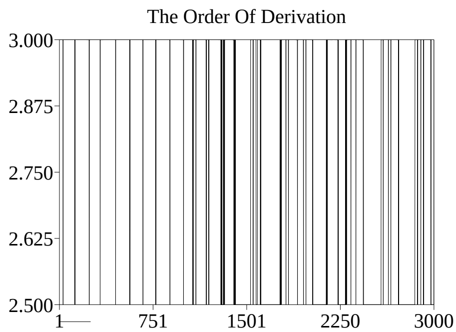


Fig. 5. The variation of the order of derivation vs. time in ms in the case of original (upper figure) and the increased (lower figure) viscosity

[5] Tikk, D.—Biró, G.—Gedeon, T. D.—Kóczy, L. T.—Yang, J. D.: Improvements and Critique on Sugeno and Yasukawa's Qualitative Modeling. IEEE Trans. on Fuzzy Systems, Vol. 10, 2002, No. 5, pp. 596–606.

[6] Kóczy, L. T.—Muresan, L.: Interpolation in Hierarchical Rule-bases with Normal Conclusions. Nikhil R. Pal, Michio Sugeno (eds.), Advanced in Soft Computing-AFSS 2002, AFSS International Conference on Fuzzy Systems, Springer, Calcutta, India, 2002, pp. 34–39.

[7] Tar, J. K.—Rudas, I. J.—Bitó, J. F.: Group Theoretical Approach in Using Canonical Transformations and Symplectic Geometry in the Control of Approximately Modeled Mechanical Systems Interacting with Unmodelled Environment. Robotica, Vol. 15, 1997, pp. 163–179.

[8] Tar, J. K.—Bitó, J. F.—Kozłowski, K.—Pátkai, B.—Tikk, D.: Convergence Properties of the Modified Renormalization Algorithm Based Adaptive Control Supported by Ancillary Methods. Proc. of the 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.

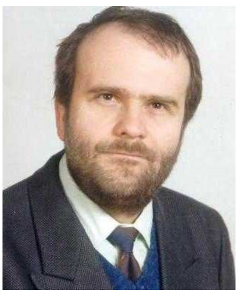
[9] Tar, J. K.—Bencsik, A.—Bitó, J. F.—Jezernik, K.: Application of a New Family of Symplectic Transformations in the Adaptive Control of Mechanical Systems. In Proc. of the 2002 28th Annual Conference of the IEEE Industrial Electronics Society, November 5–8 2002 Sevilla, Spain, Paper SF-001810, CD issue, ISBN 0-7803-7475-4, IEEE Catalog Number: 02CH37363C.

[10] Bröcker, M.—Lemmen, M.: Nonlinear Control Methods for Disturbance Rejection on a Hydraulically Driven Flexible Robot. In 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.

[11] Tar, J. K.—Bröcker, M.—Kozlowski, K.: 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.

[12] Tar, J. K.—Rudas, I. J.—Bitó, J. F.—Tenreiro Machado, J. A.: Fractional Order Adaptive Active Vibration Damping Designed on the Basis of Simple Kinematic Considerations. Proc. of the 2nd IEEE International Conference on Computational Cybernetics (ICCC '04), August 30–September 1, 2004, Vienna University of Technology, Austria, pp. 353–357, ISBN 3-902463-01-5.

[13] Kröell Dulay, I.—Fűrész, F.—Harkay, G.—Lukács, J.: Fundamentals of Hydraulic Power Transmission. (Ed. I. Kröell Dulay), Budapest, Hungary, Vol. 7, 1988, pp. 38–44; pp. 99–100, ISBN 0-444-98973-0.



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